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SEASONAL VARIATIONS IN THE FLOW OF OIL WELLS

By R. TOMASCHEK *

SUMMARY

It is shown that certain flowing oil wells in the Eakring area (Nottinghamshire, England) which have not otherwise been disturbed display annual variations in the flow of total fluid. The investigation has been extended further to show that a similar periodicity exists in the production of a whole oilfield (Kelham, Nottinghamshire) which has not been disturbed unduly by outside influences. There appears to be a definite connexion between these variations and the seasonal movement of the earth's crustal layers as evidenced by observations of the tilts of the earth's surface and by the correlation to the periodicity of shallow earthquakes.

I. INTRODUCTION

IT is well known that the crust of the earth is not a rigid shell but that it displays movements of its own. These are partly due to tectonic and geological forces, but there exist other movements of a periodic character due to gravitational influences of the sun and the moon producing tidal effects in the solid crust.[1] Furthermore, there exist movements of more or less periodic character which are effected by meteorological conditions like the seasonal changes of temperature or pressure distribution. These movements of the earth's crust can be observed even at a greater depth, showing that larger areas are involved. They can be detected by instruments, such as the horizontal pendulum, which are able to register the exceedingly slight tilting of the earth's crust connected with these movements.

In the course of such observations, which will be reported in detail elsewhere, it was found that there exists a seasonal period in these tilts. The experiments were performed at a depth of 450 ft underground in the I.C.I. salt mine at Winsford, Cheshire, with two horizontal pendulums in two perpendicular azimuths. Due to the considerable depth, influences of temperature and disturbances on the tilt by rain, storms, frost, etc., could be entirely eliminated. A discussion of the results shows, furthermore, that the elastic hysteresis of the instruments is negligibly small. Fig 1 which represents a part of the observations, can therefore be regarded as giving a reliable picture of local tilt in the upper strata of the earth's crust. It shows that during the summer the crust is very quiet and steady, but in autumn and spring, there are periods of surprisingly strong tilting. There is no doubt that the main features are due to the change in energy flow from the sun to both hemispheres of the earth's globe. There is some evidence that these sudden tilts can be correlated with the annual change in acceleration of the earth's rotation, and are possibly released by the strain and stresses which are generated by this change between the interior and the

* Anglo-Iranian Oil Co. Ltd.

surface of the globe. These problems will be dealt with in detail elsewhere. It may be borne in mind that though this curve is typical in its main features because of its primary origin in strictly periodic astronomical causes, the details of its shape may vary just as meteorological conditions change in different years, so causing the variability of the general character of a season over several years. As far as local conditions are concerned,

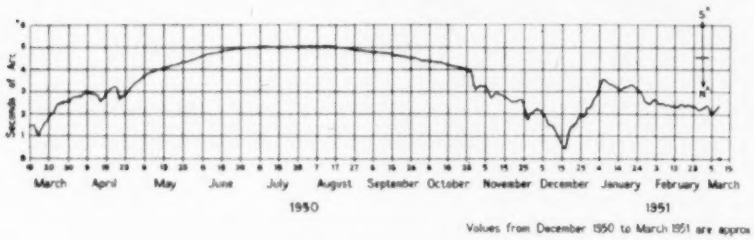

FIG 1

HORIZONTAL PENDULUM—SEASONAL TILT AT WINSFORD—AZIMUTH : S 20°E

there has to be expected also the influence of such local peculiarities as relative position of faults and main tectonic structures.

II. PROBLEM OF THIS PAPER

As a consequence of the alternation of these quiet and agitated states in the strata of the earth's crust, it could be expected that deep wells, which are not influenced by the meteorological conditions on the surface, should show a minimum flow during the summer months, when the strata are not especially disturbed. As the output in oil wells is usually measured fairly accurately, it seemed advisable to investigate if such influences could be found in the wells of the Eakring areas as these fulfil, to some degree, the conditions necessary for such an investigation :

first, there are available carefully determined measurements of the flow of the oil wells;

secondly, there is sufficient material obtainable in regard to oil wells which flow under natural undisturbed conditions;

thirdly, the flow is not restricted by interference from outside (for example, no restriction of output according to demand).

It is obvious that every well will not show the influence sought in the same way, even if the conditions of its flowing without outside interference are fulfilled, but, as the results given later show, this influence seems to be quite widespread in this area.

III. OBSERVATIONS ON SINGLE WELLS

(a) *Material*

First, the data were selected of two wells from Eakring, No. 10 and No. 101, which were very little interfered with, and the years with practically

no outside disturbance were chosen. The depth of the wells is about 2000 ft.

The data used in the following discussion are those of the total daily flow (oil plus water). This was determined at these wells at weekly to fortnightly intervals by pumping them for 24 hr to the basic pump level into a fixed reservoir or a tank wagon. These numbers characterize therefore the total fluid which could reach the bottom of the well out of the surrounding strata during 24 hr. As the wells were pumped continually in the meantime, this gives the maximum possible output under fairly constant conditions as far as the possibility of accumulation of fluid near the bottom of the well is concerned.

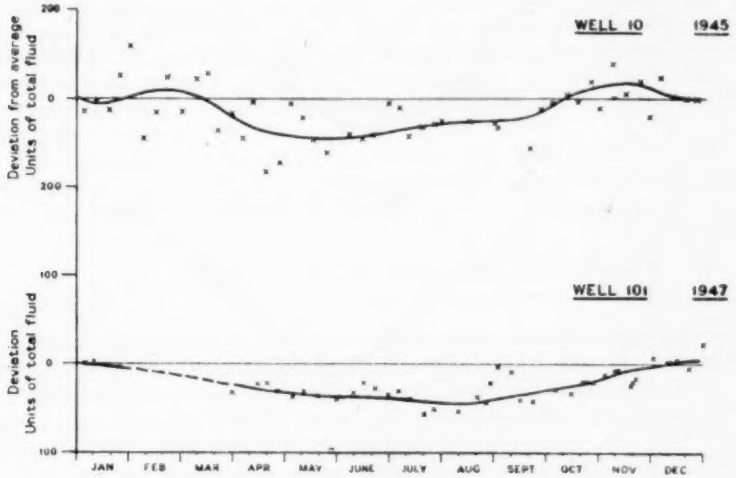

FIG 2

VARIATION OF TOTAL DAILY FLUID (OIL AND WATER) AT EAKRING—DRIFT ELIMINATED

The reliability of the single measurements can be shown by comparing measurements on subsequent days, as for this short time the conditions in the strata can be regarded as constant.

Such measurements showed on Well 10 and 101 in gallons :

First day	764	241	235	228	228	212
Second day	757	241	237	235	228	217
Difference	7	0	2	7	0	5
Difference per cent . .	0·9	0	0·9	3·0	0	2·3

The mean deviation is $\pm 0{\cdot}6$ per cent, which can be regarded as representative for these measurements. As each point of the curves shown later is the mean of several measurements, the standard error of these points can be estimated to be less than $\pm 0{\cdot}2$ per cent as far as the insecurity due to measurement is concerned.

(b) *Results*

As the productivity of the wells declines with time, the data have to be reduced to be comparable. This has been done in two independent ways.

The first method was graphical. The drift curves were taken by graphically interpolating the monthly averages. (See also Section IV (*b*).) Fig 2 shows two examples containing the measurements in gallons on each well over a special year after the drift had been removed in this way. The crosses show the measured values and give an impression of the statistical scattering. The average curve is also shown in this figure. It shows

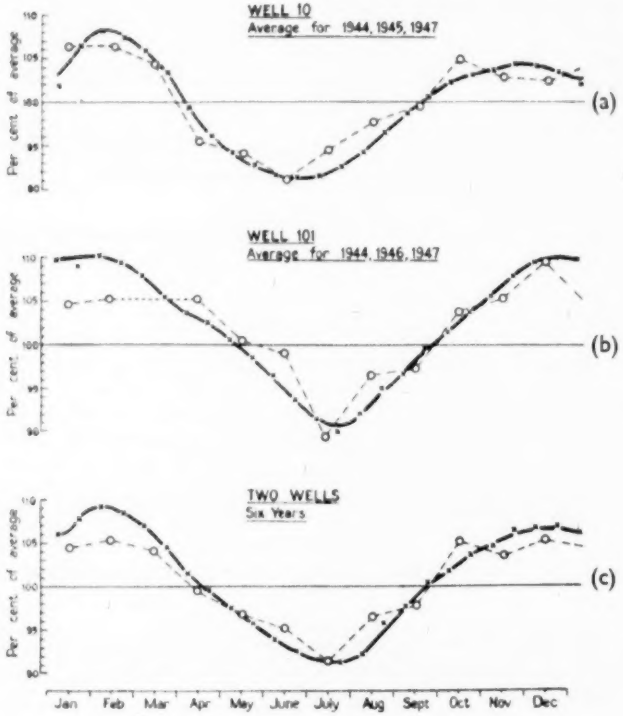

FIG 3

MEAN VARIATION OF TOTAL DAILY FLUID AT EAKRING—DRIFT ELIMINATED

distinctly the expected effect of a minimum yield during the summer months. To make sure that this behaviour is not accidental, all years in which these two wells were not disturbed were investigated. The smooth curves connecting the crosses in Fig 3(*a*) and (*b*) show the seasonal variation of the average over the years for the individual wells, and Fig 3(*c*) presents it for the average of both wells together, indicating that the effect is identical for both wells.

The second method was the following. Using periods over as long a

time as possible, the half-yearly averages of the experimental values were interpolated by linear logarithmic interpolation, as the curves of decline of yield are very nearly exponential. The mean monthly values obtained in this way were compared with the monthly averages of the experimental values. The results are shown in Fig 3 by the circles connected with dotted lines. Both methods give very good agreement within the limits of their different procedure; *e.g.*, the monthly averages in the second method not taking into consideration the distribution of the single values within the month. The second method shows somewhat lower values for the first part of the year compared with those obtained by the first method. This is due to the circumstance that in the second method the monthly values are compared with the monthly averages according to a logarithmic law, whereas in the graphical method, in order to avoid the introduction of any assumption, the fortnightly values are compared with the annual average, which procedure enhances the values in the first part of the year. It seemed advisable to retain this feature in the curves to show that the assumptions on the special law of drift do not influence the showing of the effect significantly.

The reality of the effect is shown by the very close agreement not only in the behaviour of both wells, but also with the curves obtained in Section IV. See also Section VIII.

IV. Observations on Total Oil Production

(a) *Problem*

As the effect appeared to be quite distinct for certain individual wells, it seemed worthwhile to investigate if it could be seen in the oil production of an entire oilfield which had not been essentially interfered with from outside. The Kelham oilfield, which lies about 6 miles south-east of the Eakring field, presented such an opportunity.

For various reasons it is not to be expected that the effect should come out so clearly in the production curves. First, the ratio of oil to water is not quite constant, and the oil-production figures can give, therefore, only an approximate picture of the total flow of fluid (oil plus water). Secondly, a certain amount of interference (clearing casing, etc.) is unavoidable. As these operations are fairly equally distributed over the year, they will not introduce another periodicity, but will tend to equalize the curve and therefore diminish the amplitude of the effect sought for. Thirdly, there could be, as already mentioned in Section II, an influence of production needs on the output (*e.g.*, diminishing the output with diminishing demand). This had not taken place at these wells, so that the figures are free from such an influence. Furthermore, an influence on production possibilities due to outside circumstances is to be expected. For example, lower production should be expected during a severely cold period, when the increase in viscosity of the oil and the precipitation of wax cause difficulties.* This influence is indeed quite marked in the production figures. The values are, throughout, below average in January and February, the months when severe frost is most likely to occur. On the other hand, the production

* This influence was not effective in the measurements reported in Section III.

figures for March are often substantially higher than the average. This may be due to an increased accumulation of oil in the reservoir strata during the time of restriction of outflow. But this minimum in winter and maximum in March is also expected to be a genuine effect of earth movements, as these are small in February and strongly increasing during March (Fig 1).

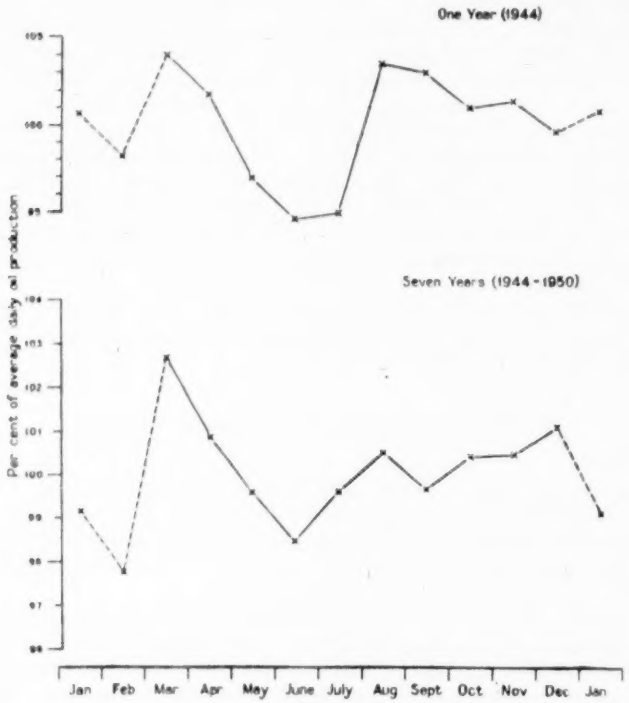

FIG 4

MONTHLY AVERAGES OF DAILY RATES OF OIL PRODUCTION OF 54 WELLS AT KELHAM—DRIFT ELIMINATED

See also Section VIII, which shows a similar behaviour for shallow earthquakes, where no influence of temperature is to be suspected.

(b) *Material*

The production data used were obtained from fifty-four wells in Kelham oilfields. Data covering seven years were available. The monthly averages of the daily rate of oil production were taken as the characteristic variable. Regarding the points mentioned above, these values give a fair picture of the mean flow towards the bottom of the wells except, perhaps in part, for January, February, and March.

As the production of the wells is not constant, the drift had to be elimin-

ated. This was done by constructing four smooth curves using the January values and respectively the March, June, and September values. The deviation of the observed values against the interpolated values of each of these curves has been determined. The observed average monthly values were set in relation to the average monthly values determined by the interpolated curve. The agreement of the single curves of the annual variation based on any one of these four curves is very good. The mean error of the single monthly value of the curve shown in Fig 4, due to the interpolation, is only 0·16 per cent. The total amplitude of the effect is thirty-five times this value. See also the detailed discussion in Section VIII.

(c) *Results*

Fig 4 shows in the upper curve the monthly averages of the daily rate of output of the total oilfield for 1944. It shows the minimum in January and February, which is expected to be enhanced by mechanical difficulties in production. Furthermore, it shows quite distinctly the summer minimum.

The lower curve shows the average over seven years. Even such an extended average shows the geophysical effect quite clearly. As with progressive years the wells had increasingly to be interfered with from outside (cleaning, casing, etc., but no shooting, acidizing, or secondary recovery), it is not surprising, as already mentioned, that the amplitude of the effect is smaller than for the first year (note the different scale of the ordinates) and, of course, smaller than for special selected wells as shown in Fig 3.

V. Discussion of Results

The following question arises : is the explanation of the observed effect by a seasonal variation in the movement of the surface strata the only possible one and if so, is it sufficient ?

The curve as given in Fig 1 can be regarded as typical and supported by other observations, as will be shown in a future publication. The basic assumption can be therefore regarded as sufficiently established to serve as an explanation of the observed effects. See also Section VII.

Are there other possibilities which may give rise to a seasonal variation of the flow ? In Sections III and IV it has already been pointed out that no reasons can be found arising from measurements or production possibilities on the surface, which could have an influence in the direction of the observed effect. Which possibilities for such an influence exist in the oil-bearing strata ? As these wells have not been interfered with, this influence must be located entirely in the strata themselves.

(a) *Influence of Rain*

One possibility which has to be discussed is the existence of variations in the water level due to such surface meteorological influences as periods of rain and similar causes. It is very improbable that the variations in the water level near the surface, which may vary periodically by one or two feet, would influence the flow in a region 2000 ft below and protected by an

impervious layer above; the geological evidence is very strong against such an explanation.*

(b) *Influence of Subterraneous Streams*

In an attempt to explain the abnormally high temperature gradients in the Eakring oilfield area, Bullard and Niblett [2] have advanced a theory that there is a flow of water from the Pennines eastwards towards Eakring through the strata immediately above the cap rock. This water accumulates in the syncline at the foot of the Eakring west flank, rises up the flank, and migrates across the Eakring–Kelham uplift towards the east. During its progress up the flank and across the Eakring area, it dissipates some of its heat and so gives rise to the high temperature gradients.

If such a stream exists, the possibility arises that it may be a cause of the seasonal fluctuations in the flow of the oil wells, either through a mechanical or a thermal agency which is itself subjected to a seasonal periodicity.

The possibility can be visualized of such a stream having an influence on the well production as a result of, say, direct connexion between the two systems via fissures or faults, but it is unlikely that it could cause seasonal fluctuations. Bullard and Niblett have estimated from the heat flow that the velocity of the stream is of the order of 30 cm/day, *i.e.*, 100 metres or so per year. The point of entry of the water at outcrop in the Pennines is 20 km from Eakring, so that its migration over this distance takes a time which can be measured in centuries. It seems inconceivable that such a stream, which at source probably has a seasonal periodicity impressed upon it, could retain this periodicity throughout its length when flowing against so considerable a resistance over this long period of time.

However, it is not impossible that the movement of this water can be affected by the seasonal movements of the earth's crustal layers. The location of this water in a layer sandwiched by two other layers which may be subjected to slightly different movements could, perhaps, enhance the effect. The result might thus be that periodically the flow of warmer water from depth to the high elevation over the oilfield would be eased, so setting up corresponding temperature changes which would affect the oil flow by lowering its viscosity.

For a liquid flowing through a capillary tube, a 10 per cent change in viscosity would account for a 10 per cent change in total flow, and to account for the fluctuations in the oil flow observed a change in temperature of about 2·5° C in the neighbourhood of the foot of the well would be necessary. This seems rather high.

There is a further reason which renders an explanation along these lines somewhat improbable. On the basis of Bullard and Niblett's theory, the thermal influence of this stream at Eakring, where it has risen locally through a considerable distance up the west flank of the structure, should be

* Such conditions may be effective in certain oil wells in Persia, where a direct connexion between the flow of water on the surface with a driving-water level below the oil is suspected and some correlation between water-drive and rainfall seems to exist. But even there it has been observed in a certain well that the oil/water level rose even at times where no explanation due to rainfall can be given. The times of increased oil/water level in this case coincide with those which would be expected by the influences discussed here, so that the explanation by earth movements may to a certain extent also be applicable in this region.

very much greater than it is at Kelham, which is 6 miles or so farther south-east and at about the same elevation as Eakring. But Figs 3 and 4 demonstrate that the seasonal change of flow of the wells is observable at both places, and is of the same order of magnitude under undisturbed conditions, when account is taken of the " averaging effect " of having considered the whole oilfield and oil production only in the case of Kelham.

(c) *Barometric Effect*

There is also the possibility that the flow could be influenced by changes in barometric pressure. Such changes are transmitted to the bottom of the hole, and in a field with a gas cap, and with fairly free connexion throughout the reservoir, there could be quite an appreciable change in the oil flow. In the case under consideration, however, no gas cap exists and connexion is not particularly free. Supposing, however, that connexion is free enough to allow the long-period barometric pressure variations to become effective throughout the reservoir, then under the pumping conditions which exist (see Section III (*a*)) variations of flow could be expected in accordance with the resultant long-period changes in the differential pressure acting in the reservoir. Now the amplitude of the yearly period of change in barometric pressure in the Eakring area is about 0·005 atm, and this is less than one-thousandth part of the actual pressure differential acting in the reservoir in the neighbourhood of the foot of the well. Thus an annual variation of about 0·1 per cent in the output of a well may be expected as a result of changes of barometric pressure. The variations actually observed, however, are nearly 200 times greater (see Fig 3). Thus any idea of barometric pressure changes causing these seasonal changes may be excluded.

(d) *Conclusion*

From the above considerations it seems reasonable to assume that the mechanical stresses and strains set up in the crustal layers of the earth's surface, as a result of their seasonal changes in tilt, can have a direct influence on the production of an oilfield.*

VI. Energy Considerations

The question arises as to whether the energy developed by the movements of the earth's crust is sufficient to influence the flow of fluid through the reservoir strata to such an extent as is necessary to explain the observed effect. The average flow in the reservoir is dependent on pressure gradient, porosity, and viscosity (regarding the horizontal component). These values determine mainly the energy gradient characterized by the slope $tg\alpha$ of the plane of energy as shown in Fig 5, which gives a small section of this plane for a volume in the reservoir. As the porosity is determined by capillaries of a statistically distributed size, and as the oil emulsion contains a statistically determined size of droplets, the influence of capillarity will

* It could be expected that this influence is especially enhanced in the neighbourhood of faults. Although such a connexion seems indicated for one of the wells treated in Section III, the geological micro-structure of the oil-bearing strata is not known sufficiently to draw this conclusion with reliability.

be a " roughening " of the energy surface as indicated in the figure. This means that a moving droplet may be trapped in an energy hole (see Fig 5), where it remains if there is not an additional energy *dE* available to bring it over the " wall " of this hole. This energy will be only small, and the statistical fluctuations of energy may be sufficient, if the hole is very shallow, to free the droplet. But there may be deeper holes where additional external energy is necessary. The particle may, for example, be freed from the hole when a sufficiently steep front of a shock or vibrational wave travels along the energy surface and tilts it by passing this point.* It is obvious that the " *roughness of the energy gradient* " is a very characteristic quantity for a given oil-bearing structure, as it gives a measure for the

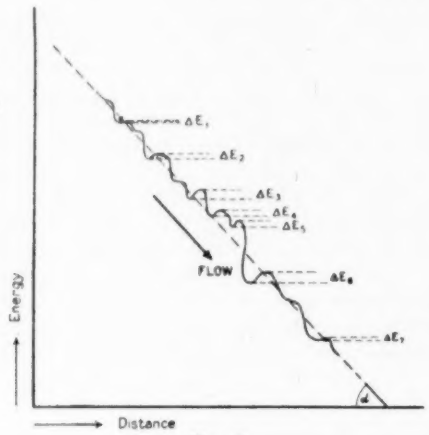

FIG 5

ENERGY DIAGRAM OF CAPILLARY TRAPS

necessary magnitude of fluctuations of energy sufficient to release a certain number of arrested particles from their traps.

As the particles move in the direction of the energy gradient every liberation, even by slight increases only of energy (of the order of magnitude of the " roughness " of the energy gradient), *increases* the flow of the emulsion towards the collecting point.

It is evident from this formal picture that even small quantities of energy of the order *dE* available from the earth movements can be sufficient perceptibly to increase the fluid flow.

A mechanical analogy is given by the following picture : a flat piece of metal is lying on an inclined plane, and the inclination is of such an angle that the metal is just not moving. Then even very slight periodic tilts or oscillations will cause the metal to slide down so long as these movements continue. The amplitude of the necessary tilt will depend on the roughness of the surfaces.

* For the same amplitude of energy therefore vibrations of higher frequency should be of advantage for this sort of liberation, as their front is steeper. Therefore an influence of microseism may be expected.

It is difficult to estimate the possible amount of energy developed by the seasonal tilting of the earth's crust before definite knowledge is available of the real distortions caused by them.

As far as quantitative calculations are concerned, some [3] consider the volume change in the surface layers due to compression and dilatation by the tidal forces. The result is that the dilatation $\Delta(a)$ near the surface has the same sign as the radial displacement Δh and that

$$\Delta(a) = 0{\cdot}323\,\frac{\Delta h}{a}$$

where a is the earth's radius at the surface.

The simplifications used do not seriously influence the result for this application. The dilatation is therefore of the order of magnitude of the ratio of the tidal displacement to the radius of the earth. This amounts to about $1{\cdot}5 \times 10^{-8}$ for the lunar tide.* It means, as C. L. Pekeris [5] shows, that a change in level of 1 cm in a well of 1 metre radius could be brought about by the above dilatation in a water-bearing hemisphere of only 100 m radius. A cavity of 1 cubic metre filled with liquid, and sealed by a capillary would undergo a volume change of the order of 10 cubic millimetres, an easily observable effect.†

In Fig 1 the tidal oscillations of tilt, corresponding to these figures of change in volume, are so small, that they are scarcely visible, as they are approximately of the thickness of the line. The irregular, more local tilts, held responsible for the variations in the flow of the oil wells and represented in Fig 1 are ten to a hundred times greater in amplitude. As the phenomena connected with these irregular tilts, due to their origin, should be confined to a comparatively thin layer of the earth's crust, it can be expected that their influence is, at least locally, several orders of magnitude higher than the tidal effects calculated above, especially if the high shearing stresses which may occur in certain regions are taken into account.

VII. Correlation with Seasonal Variations in the Frequency of Earthquakes

If the reason for the observed periodicity in oil wells is to be found in internal movements of the oil-bearing layers, there should exist a correlation to other phenomena connected with such movements. One obviously to be connected with these movements is the release of earthquakes.

The question of seasonal variations in the frequency of earthquakes is not yet settled.[6] This is not surprising, as a yearly periodicity may only contain a sort of " trigger " effect and therefore give only a periodicity which is superimposed over a large number of statistically evenly distributed events. Since the cause of the variations treated in this report is to be found in the uppermost layers of the surface, it is to be expected that shallow earthquakes (depth less than 60 km) should be best correlated to the observed periodicities. Furthermore, statistics of smaller earthquakes

* Direct measurements with a special strainmeter in a mine at a depth of 800 m and in a tunnel confirm the existence of these tidal strains and their amount.[4]

† N. O. George and F. E. Romberg observed such tidal variations in an Artesian well (*Trans. Amer. geoph. Union*, 1951, **32**, 369–71).

should show such a correlation more clearly, as the energies necessary to " trigger " them may be smaller, and the periodicity may become clearer by confining the data to a smaller area with more constant conditions.

There exists a very thorough examination over fifteen years of 676 earthquakes in Greece.[7] Fig 6(*a*) shows the yearly variation of frequencies. It is surprisingly well correlated to the curves discussed in this paper.

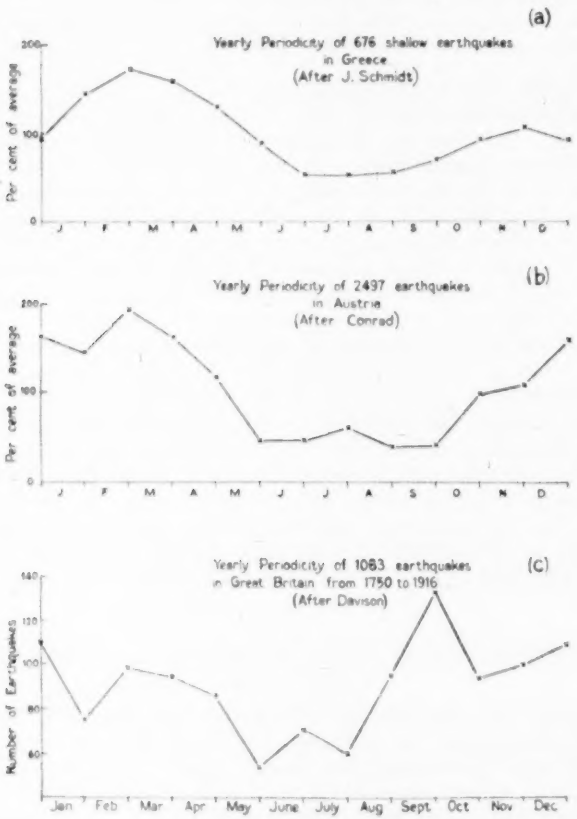

FIG 6

PERIODICITY OF EARTHQUAKES

The same type of behaviour is obvious in Fig 6(*b*). It shows the yearly periodicity of earthquakes in Austria according to a very careful and critical study by Conrad.[8] He shows also that the annual period for Norway, Switzerland, and Austria is very similar. The minimum in Norway is about July 8. As Norway is connected with Great Britain by the Caledonian structure, where also Winsford is embedded and which extends below the oilfields here under consideration, this parallelism has geological support.

Even more convincing is Fig 6(*c*), showing the yearly periodicity of the earthquakes of Great Britain from 1750 to 1916,[9] as it concerns the same region here under consideration.

The strong correlation between the curves of Fig 6 to the curves of flow given in this paper and the evidence given above shows that the seasonal variation in oil wells may be connected with the same conditions which are able to enhance the release of earthquakes. And the obvious correlation of both types of curves to the seasonal variation in tilt given in Fig 1 makes it very probable that actual movements of the earth's crust, which by themselves are correlated to meteorological or geomechanical processes, are the common cause as will be shown in due course elsewhere.

VIII. Appendix—Harmonic Analysis

One of the criteria for the reality of the observed periods is Schuster's.[10] For the set Kelham 1944 the expectance is $\epsilon = 0{\cdot}014 = 1{\cdot}4$ per cent; this means that for the number of observations made, this would be the mean amplitude of a purely statistical distribution. Applying this to the corresponding curve in Fig 4 the overall amplitude of this curve is about ± 5 per cent, which is a significant multiple of the expectance, showing the reality of the period. Furthermore, to check this point by determining the standard errors of the amplitudes, they were subjected to a harmonic analysis by the method of least squares (Fig 7).

The curve Kelham 1944 can be represented in its annual and semi-annual constituents in per cent by

$$\Delta H = 1{\cdot}97\ (\pm 0{\cdot}58)\cos(x - 317^\circ) + 2{\cdot}86\ (\pm 0{\cdot}41)\cos(2x - 125^\circ)$$

Similar analysis of the curve of British earthquakes yields in per cent

$$\Delta H = 21{\cdot}4\ (\pm 4{\cdot}4)\cos(x - 330^\circ) + 17{\cdot}5\ (\pm 5{\cdot}6)\cos(2x - 190^\circ)$$

The minimum of the annual period of Kelham 1944 is at June 2 and at June 28 for the Kelham seven-year period. If the weight 1 is attributed to the 1944 curve and the weight 3 to the seven-year curve, taking into account on one hand the longer duration and on the other the increasing disturbances a mean of June 21 is obtained. This compares with June 15 for earthquakes in Britain. The difference is entirely within the range of uncertainty of phases. The maxima of the semi-diurnal period are for Kelham 1944 at March 20 and September 20, for the seven-year period Kelham at April 6 and October 6. The mean taken as above yields April 3 and October 3. The corresponding data for British earthquakes are April 21 and October 21. There seems to exist a sufficiently significant difference between the semi-annual period. Besides differences in the mechanism, this may be partly due to the possible influences of working conditions on the January to March values as stated in Section IV (*a*), which may also enhance the semi-annual amplitude to a certain degree, as this influence is in the same direction as the influence of the earth movements. The conformity of oil flow and frequency of earthquakes is as good as can be expected taking into account the possible influences of local geologic structure.

The amplitudes are in both cases for both constituents significantly above their standard errors, and the relative ratio is in both cases of the same order

of magnitude. Furthermore, there exists a significant correlation of the phases with quite independent analogous periods. For the earthquakes there are very similar curves found for Norway, and also for Austria and Switzerland; for the oil wells we have the quite independent curves of

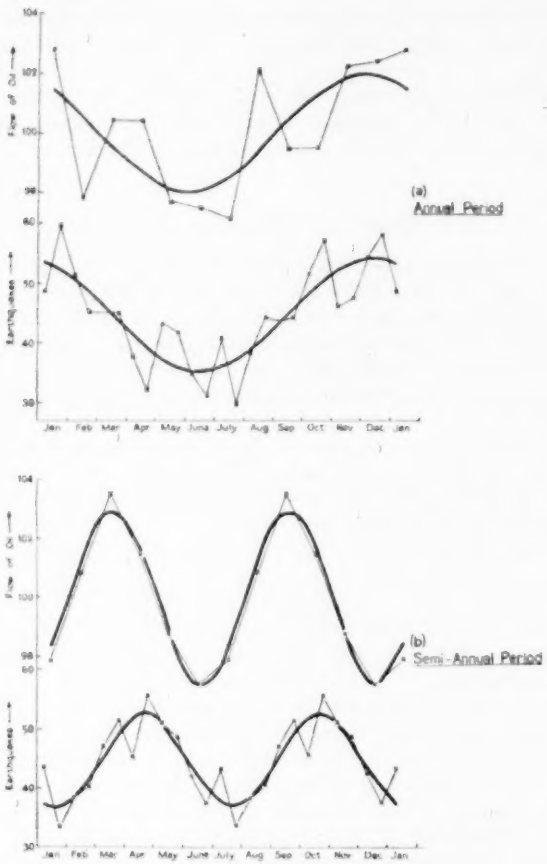

FIG 7

ANNUAL AND SEMI-ANNUAL HARMONIC CONSTITUENTS OF FLOW OF OIL (KELHAM 1944) AND BRITISH EARTHQUAKES

total flow at the two Eakring oil wells. This equality of phases increases the probabilities of the observed variations in the flow of oil wells to about one to several thousand against the occurring of these periods by mere chance fluctuations.

A numerical comparison with the curves of the earth tilt (Fig 1) is difficult, as the flow effect depends not only on the amount of strain itself, developed in the strata, but also on the oscillations superimposed on it.

The first may be approximately represented by the amplitude of the tilt observed. It shows mainly an annual periodicity with an extremum in July. The superimposed oscillations are given by the ruggedness of the curve Fig 1, which can be judged by the results of harmonic analysis of appropriate equal parts of the curve. This shows also a significant semi-annual period. The correlations found in this way are also fairly obvious from Fig 1. They give evidence that the assumed connexions with mechanical stresses which led to this investigation may indeed provide the explanation of the observed phenomena.

Acknowledgments

The author is indebted to the chairman of the Anglo-Iranian Oil Co. Ltd. for permission to publish this paper. Thanks are due also to Mr C. M. Adcock at Eakring, who very helpfully provided the necessary oilfield information, and Mr P. Docksey, Dr D. T. Germain-Jones, and Dr G. M. Lees for stimulating discussions.

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THE INSTITUTE OF PETROLEUM

GEAR LUBRICATION SYMPOSIUM

An Ordinary General Meeting of the Institute of Petroleum was held at 26 Portland Place, London, W.1, on February 13, 1952. Professor F. Morton (Vice-President) occupied the Chair during the early part of the proceedings.

His Late Majesty King George VI

The occasion being the first meeting of the Institute since the death of His Late Majesty King George VI, the meeting paid silent tribute to his memory.

The Chairman (Professor F. Morton): This afternoon and this evening we are holding a symposium on the lubrication of gears and the testing of gear lubricants. We are very happy to see so many members and guests present. It is my privilege to ask Dr C. G. Williams, the chairman of our sub-committee on the Mechanical Testing of Lubricants, to occupy the chair. It is the customary procedure in the Institute, when symposia are held on specialized subjects, to choose some person of eminence in the field in question to preside and to guide the discussion. For this particular symposium I am sure you will agree that, with a world-wide choice, we could have made no happier one (hear hear).

Dr Williams is well known to you. Before becoming the Director of the Research Centre at Thornton he was with the Motor Industry Research Association; and at one stage of his career he almost became academical. The Universities' loss is industry's gain; but if he had turned to academical paths he would have been equally famous. He is now one of the leading engineers of our industry.

It gives me very great pleasure, Dr Williams, to ask you to occupy the Chair.

Dr C. G. Williams, occupying the Chair, said: I think the large attendance this afternoon is a very happy augury for the success of our discussions.

I would like to refer to the origin of this symposium. I am chairman of the Mechanical Tests of Lubricants Panel, which for some years has been considering mechanical tests for gear lubricants. Conspicuous amongst these test methods has been the IAE machine, because it was available in a number of laboratories. An extensive programme was organized by the Panel using this machine, and has resulted in some degree of standardization of a test method which now seems fairly satisfactory.

We also considered other test methods, but we arrived at the stage at which we were not quite sure what to do next. I do not suggest that when-

ever we are in that predicament it is necessary to organize a symposium; but in the present instance we felt that the subject should be thrown open for discussion and others asked to give their views on the fundamentals of lubrication and on the testing of gear lubricants. Very briefly, that is the reason why we are here to-day.

We have two sessions. The first will deal more particularly with the fundamentals of gear lubrication, with hydrodynamic theory, and with what happens when lubrication breaks down and we have pitting, scuffing, and wear. At the second session we want to discuss the more practical aspects of testing, to consider test methods, whether we should be using the IAE machine, the disk machine, the four-ball machine, full size back-axles, or any other means of testing.

At each session a digest of the papers will be presented and Dr. Clayton has very kindly agreed to summarize the papers at this afternoon's session.

PART I. THE LUBRICATION OF GEARS

DR D. CLAYTON, presenting his summary of the papers in this Part, said:

Introduction. The best kind of lubrication is that which keeps the mating surfaces apart altogether, viz., hydrodynamic lubrication. Then there is no wear, apart from the effects of particles; also the friction is lowest under these conditions. When the surfaces touch we are at the beginning of the danger zone; it is important to remember in our enthusiasm for studying what happens during contacts that it is really establishing the limit of safe operation so that the user can avoid the danger. It is logical therefore to consider these papers in this order of subject matter: hydrodynamic lubrication; separation of the surfaces by other means—in this case the one paper on colloidal graphite; then what happens in touching under moderately severe conditions; and finally, ultimate surface failure by scuffing. That is my basic plan.

You will have noticed how much work has been done with disks in edge contact to imitate gear-tooth operation; no one has taken on the task of establishing the case for disks in his paper. Moreover, there is vague reference to " gear teeth " without making clear in every case what kind of gear is being considered: the difference between a spur gear and a worm gear is considerable, and there is a notable difference between hypoid-gear-tooth action and that of spiral-bevel gear teeth—a difference which, if properly analysed some years ago, could have saved much time in research.

The worm-gear conditions are more closely simulated by constant-speed disks than are spur or helical gears, because in the latter there is considerable variation of speed of sliding and a momentary rolling action at the pitch line. For some years there has been a design of disk machine that allows cyclic variation of speed to give more exact simulation, but Cameron indicates another ingenious design to achieve the same thing by using the face of one disk against the edge of the other. However, for his hydrodynamic theory he goes back to the edge-on contact. Smith also refers to a variant of the disk machine in which the disk axes are slightly

askew, to give point contact, and then, by slight axial movement, a sinusoidal path of contact is produced. (However, I cannot see from his description how that imitates gear-tooth action.)

Before leaving disk simulation it is worth pointing out that it is more difficult still to imitate hypoid-gear-tooth action by disks, particularly because the motion of the disks is at right angles to the line of contact, whereas in the hypoid gear there is motion along the line.

Hydrodynamic Theory. There are two papers on hydrodynamics, one by Cameron and the other by McEwen. In the first Cameron works up the formulæ for the pressure in the film, the load carried by it, the coefficients of friction, and the energy dissipated in the film, all for constant viscosity of the oil. He shows that the film thickness increases with increase of viscosity, speed, and relative radius of curvature, and decreases with increase of load, in the same way as with journal bearings with which some may be more familiar. He makes the important qualification that the speed cannot be increased without a decrease of the viscosity. He says that the theory predicts a load much less than can actually be carried. The film thickness will, of course, inevitably be thin with the opposing curvatures, and he gives a figure for one experiment of $5\frac{1}{2}$ millionths of an inch.

Cameron then takes up the problem of increasing viscosity with pressure, but this case is treated more fully by McEwen. It is clear that the effect of pressure on viscosity can help considerably in building up a fluid film. The film pressure rises to infinity for a certain film thickness for which the formula is given, and to McEwen this represents the limit below which fluid film lubrication cannot go; if the roughnesses are larger than this minimum film thickness they limit the fluid film régime. Cameron says it means the disks become oil-borne whatever the load.

Cameron further considers temperature effects, and then the influence of metal deformation; by locally increasing the radius of curvature, or making flats, this can increase the load-carrying capacity. He mentions also the film formation due to roughness of the surfaces, including Lane's case of oil being trapped between opposing roughnesses.

From a practical point of view it is clear that increased viscosity is desirable to secure the thickest fluid film for a given load and speed, until heat effects cancel out the benefits or there is difficulty in supply. Alternatively, it is clear why the viscosity can be lowered for high-speed gears, and why it must be increased for heavy loading of low-speed gears. It would clearly appear to be desirable to choose lubricants sensitive in viscosity to pressure, but unfortunately with petroleum products such materials are also very sensitive to temperature, and it is difficult to be sure of gaining an advantage.

It seems difficult to get a low modulus for the metals without also getting a low elastic limit.

The question of surface roughness is one for debate. As the films are inevitably thin it would appear desirable to make the gears as smooth as possible, as inferred by McEwen and at one place by Cameron. There is some evidence, however, that surface roughnesses can themselves help to build up the film (they can be of benefit even near to scuffing as

Barwell and Milne show), particularly if they are at right angles to the direction of motion. We must not, in concentrating attention on roughness, forget that the geometry of the surfaces and the mounting must also be accurate.

There is no reference to what I personally regard as one of the most promising recent suggestions, viz. : Gatcombe's, that vibrations normally between the surfaces may build up film thicknesses six or so times what one can expect from normal hydrodynamic action.* This is an important effect in journal bearings, and there is scope for further experimentation with gears, and, if confirmed, a rewarding enterprise for someone to show how this benefit can be secured without disadvantage.

Separation of the Surfaces by Solid Particles. There is a short paper by Smith outlining the part graphite can play in gear-tooth lubrication. He refers to its characteristic of easy shear, enabling it to keep friction low and to spread itself over metal surfaces. He quotes a coefficient of friction of 0·10 to 0·13. He says that graphite has a high mechanical strength at right angles to the shear planes and that this, with the facility for slip in shear, enables the graphite to offset metal pick-up between surfaces. He claims a moderate increase of load-carrying capacity for graphite as an additive to mineral oil, but disclaims extreme-pressure qualities. He claims properties not always possessed by other oils such as the reduction of certain pitch-line noises and facilitating running-in with minimum tooth wear; also the ability to sustain gear lubrication with high oil-bath temperatures. Finally, he calls attention to graphited greases, but draws special attention to the need for homogeneous dispersion and for *pure* graphite.

I was rather disappointed, as there is only one paper on colloidal graphite, that Smith does not refer to the considerable amount of information that has been published in recent years on graphite. It would have been a grand opportunity for the specialist in graphite to give us a really balanced account of the present position.

Contact under Moderate Conditions. Taking the early stages of surface contact, Finch and Spurr emphasize two important points : (1) the vital role played by oxide films in preventing and impeding the inter-metallic contact that would give very high adhesion, and (2) the influence of the rigidity of the metal in governing the support and any cracking of the protective oxide layer. The protective quality of the oxide layers may be further enhanced by adsorbed gases and lubricant molecules.

The conditions are mildest at the pitch line of spur or helical gears provided the elastic limit is not exceeded in bulk. When shear due to relative sliding occurs there is a greater tendency for the protective layers to be pushed aside; base metal so exposed provides opportunity for adhesion, but also the material pushed aside may collect and form particles that cause excessive loading and plastic deformation of the underlying metal, causing further adhesion. After welding occurs, parting below the surface occurs because of work-hardening at the surface, and progressive damage results.

To keep the general picture clear I link in here the ideas of Barwell and

* *Trans. A.S.M.E.*, 1951, **73**, 1065.

Milne on the milder contacts, and relate them to the hydrodynamic condition. When contact first occurs as the duty on a hydrodynamically lubricated gear is increased, it is at odd spots. The consequences just described in the way of very local plastic deformation and heat generation occur, but the heat is dispersed in the lubricant and the material is removed from the high points; a full fluid film can in this way be restored, and this is the best kind of running-in, as Evans and Tourret deal with it. As they show further, if the conditions are rather more severe a greater amount of wear occurs and the running-in may never finish. They are concerned with the conditions that lead to continuing wear, and I will take up their story later. I want to emphasize that here we are at the boundary between the two régimes. On the one side of this boundary, the hydrodynamic, by increasing the speed the film thickness can generally be increased (though something may be lost through warming the oil more and lowering its viscosity), and therefore more load can be taken again. On the other side of the boundary increase of speed increases the heat generated at the points of contact and makes matters worse; the load has to be lowered to offset this effect. Thus speed and load stand in a different relationship on the two sides: load $\propto$ speed and load $\propto$ 1/speed. It is highly desirable to keep to the hydrodynamic side of the boundary, but highest efficiency is obtained near to it, and in the course of ordinary affairs the boundary will be crossed from time to time. It is therefore necessary to know what happens: hence the interest of these papers on failure. The boundary is not a narrow one, and contacts of the surfaces at different points can go on with some systems more or less all the time, as in Evans and Tourret's experiments; if the conditions are made too severe, however, an unstable region is met, as described later.

Thinking further of the boundary region, repair of damaged oxide films is important. Is there always enough oxygen dissolved in the oil and readily enough available? Oiliness additives can help to reinforce the oxide layers, but is enough done to ensure that there are good enough protective layers on the surfaces? I doubt it. Mansion shows the benefits of phosphated surfaces against scuffing; they can be valuable here also.

Turning now to the wear part of Evans and Tourret's paper, note their reference to the transfer of bronze to the steel in running-in, the thickness being so small that it was not detectable with the roughness-measuring instrument. They consider that there is a mixed state of hydrodynamic and boundary lubrication, the hydrodynamic not being so good because there is considerable rubbing of the disks in opposite directions. The important point they make is the *mechanism* of wear. When the load is low enough for the running-in wear to decrease to a low value, the wear may continue at a low rate for a time, but there is later an increase due to removal of a brittle layer that develops by work-hardening; then another step in the curve develops; the steps become smaller due to overlap of the processes and eventually they merge. Thus, wear is seen as a fatigue process, and the lubricant is important for two reasons: because of its viscosity it affects the load distribution and because of its lubricating properties the friction is affected, the two factors both influencing the work-hardening. It is shown that least wear occurs with high-viscosity plain oil; it is greater

when viscosity is lowered and when a S–Cl EP material is present. (Was it thought that the S–Cl additive would lower the friction?)

Severe Failure. Scuffing—both Finch and Spurr and Barwell and Milne run over the fundamental aspects that must be considered in forming a clear conception of the mechanism of failure in this way. Many of the same factors as for failure under mild conditions still hold, but the deterioration is no longer so local, the outstanding difference being that the heat developed is much greater and the temperatures are therefore higher and larger areas reach those temperatures. Adsorbed films are dispersed, and the metals ultimately soften, and this still further allows the oxide layers to be disturbed. There is now less chance of local damage remaining small and becoming innocuous; it tends to build up progressively, the system becoming unstable. The building-up progresses fastest when there are long paths of contact for any spot, and that is why the hypoid gear is so severe. There is support for there being a critical temperature on the lines proposed by Blok and Lane for any lubricant under given conditions. This is beyond the range of ordinary lubricants in the hypoid gear as ordinarily run, and the S–Cl additive-type lubricants are needed. Finch and Spurr say a surface compound layer must be formed the melting point of which lies well below that temperature at which the rigidity of the supporting metal becomes impaired, but I think they mean more than that. Unless when melted the compounds are more effective than a mineral oil, they will not serve their purpose. It is likely that at temperatures below the critical for the material the chemical action starts and, due to lowering of the friction (either by forming an easily sheared layer or by a quasi-hydrodynamic action), the critical temperature for the material is never reached; if it is, then the extreme-pressure lubricant has failed.

In general, it is clear that we can help ourselves, as Barwell and Milne say, by choosing lubricants and rubbing materials of low affinity so as to give low friction and adhesion through to high temperatures, and also by improving heat dissipation—increased thermal conductivity is not promising, but roughening in some cases can reduce the path of local heat flashes, and so also can increasing the hardness of the surfaces.

Scuffing is reached with other gears as well as hypoids, but at the severe end of a range of conditions. We turn then to Mansion's collection of results for spur gears, obtained over several years with the IAE 3¼-inch-centres machine and a series of helpers. As regards the machine it is important only to note that it works with small, narrow-faced spur gears. They are regarded as giving more reliable ratings than disks; thus better correlation was obtained with "Merlin" propeller reduction gears than any other test machine. They are closely controlled standard gears, and in use are loaded progressively each 5 minutes until scuffing occurs. The paper describes the influence of lubricants, gear materials, and running conditions on the scuffing load. It is important to note that the scuffing starts in the sliding region and does not run through the pitch region.

Now the results. Taking pairs of oils and mixing in different proportions to give a viscosity range of 15 to 160 cs at 140° F, the scuffing load increased as the square root of the viscosity, the highest being 3·2 × the lowest. This might, however, be due to the influence of the larger molecules in the

higher-viscosity oils. Variations of viscosity, caused by variations of temperature for a given oil, have been shown to have considerably less effect than equivalent differences of viscosity at a given temperature. Paraffinic oils and white oils gave very little change in scuffing load for variations of supply temperature from 40° to 110° C; one heavy naphthenic oil showed little change, but a light one showed a marked decrease with increase of temperature. Penna dark stock was better than a comparable Penna bright stock, and the indications are that Venezuelan distillates are slightly worse than blends containing Penna bright stock of similar viscosity. Also acid-refined naphthenic oil was better than a blended paraffinic, but a naphthenic oil that was highly acid-refined and solvent extracted to make it white was rather worse. Castor oil was 1½ to 3 times as good as mineral oil.

I think it can be inferred from this, though Mansion does not say so, that viscosity away from other qualities has probably little effect on scuffing. (But this, of course, does not mean that there is no effect for conditions remote from scuffing.) This may seem academic, as normal oils do vary in nature with viscosity, but it is not; for, if true, one is free to alter, by means of additives, the nature of a thin base without altering its viscosity—a very important point in some cases.

Results are given for a series of additives in an SAE 50 oil, some making the scuffing load more than three times as high. In a series of concentrations of tricresyl phosphate the asymptotic stage was being reached at 2 per cent. A series of oils to one specification gave a range of loads from 1 to 1·8, and 1 per cent tricresyl phosphate raised the loads from 1·8 to 2·4.

Take now gear materials and surface treatments. There was no systematic variation of scuffing load with surface hardnesses in the range 360 to 840 V.P.N. High nickel content was a disadvantage; both Cr and Mo were beneficial. It was not easy to make deductions on microstructure; two macrostructures were the same. Nitriding was very good, the best of the materials examined. 0·2 per cent lead for improving machinability had no effect. Combinations of different materials failed at loads intermediate between those of the individual materials. Of chemical treatments ferrous and manganese phosphating was best, doubling the scuffing load, even after prolonged running. Of electro-deposited metallic platings silver, tin, bronze, and speculum were best, about doubling the load; soft copper was next. Plating thickness did not seem to be important within the range 0·00005 to 0·0003 inch.

Speed had a marked effect, scuffing load being proportional to $1/v^n$, where v = sliding velocity and n is slightly over 1; a 1 : 4 range of speed gave a scuffing-load range of 5 : 1. Changing the lubricating-oil jet from the ingoing to the outgoing side of the gear mesh had no effect.

Barwell and Milne take up the question of the scuffing criterion, which Mansion's work shows is primarily a function of load and speed, viscosity effects being at any rate small, but oiliness qualities showing. Because it is at the tips of gear teeth and not at the pitch line that scuffing occurs, and it is thus dominantly a function of sliding speed, as already noted, Barwell and Milne have investigated the phenomenon in an apparatus giving reciprocating motion and in one giving continuous motion. The

results for reciprocating motion support the load × speed (WN) criterion, which is near to Mansion's, but for continuous motion this criterion is not as good as $\mu W\sqrt{N}$ (μ = coefficient of friction) which is suggested by Blok's temperature-flash criterion. (The latter is based, I think, on no consideration of fluid-film effects, and Barwell and Milne have demonstrated by pressure measurements a fluid film for the continuous motion; they had inferred it from the value of the coefficient of friction, and they have also shown effects to be expected with thin fluid film conditions as regards different finishes.)

Barwell and Milne then go on to give information about the changes in the surfaces during scuffing: evidence of fusion and welding—blobs of material transferred from one surface to the other for low and moderate speeds; with very high speeds of sliding, however, material is removed in small particles from both surfaces, and scuffing of the usual kind does not occur. Under the microscope the surfaces show little change until scuffing occurs, and then there is a profound change indicating the high temperatures reached and the sudden subsequent cooling; the transformed material is very hard. Disturbance to some depth has occurred.

Having considered scuffing, the other mode of severe failure is by *pitting*, discussed by Evans and Tourret. They make out a good case, with a careful collection and discussion of evidence from the literature, that the pits arise in either or both of two ways, according to circumstances: (1) shallow surface cracking can be started by the alternating compressive and tensile stresses as the disks rotate, aggravated by the fluctuating surface temperature, and by surface irregularities (leading to stress concentrations), and influenced by the method and rapidity of surface machining; (2) deeper internal cracking due to fluctuating Hertzian shear stresses, with subsequent propagation to the surface; these cracks may be brought to light at the surface more quickly if there is wear. Once these cracks reach the surface any lubricant is a disadvantage in that by hydraulic action the crack is propagated. A high-viscosity oil is better than a low-viscosity oil in these tests, and there is a disadvantage in having any EP additive present. This last result may be due to stress concentrations due to chemical attack.

The authors very sensibly consider the practical implications of their work. As the cracking is a fatigue phenomenon, maximum fatigue strength compatible with other requirements is obviously desirable. Stress raisers in the form of surface irregularities, and any bad influences from machining, should be avoided, as well as chemical effects. Viscosity and wetting properties will affect the access of lubricant to the crack, and so influence the tendency to pit. They point out that other circumstances must be considered, *e.g.*, a high-viscosity lubricant may cause too much heat development by viscous drag and lose its advantage. I do not think they mention that a higher viscosity of the oil at the contact would spread the otherwise Hertzian load over a wider area and thus considerably ease the maximum stresses. They do say that in some cases increase of viscosity will lower the friction and enable a higher proportion of the load to be carried hydrodynamically.

The papers presented at the afternoon session follow.

HYDRODYNAMIC THEORY IN GEAR LUBRICATION

By A. Cameron *

Introduction

This subject is the one that tries to answer the question " Why do gears get lubricated ? " The practical question of " how " is subsidiary at the moment, though an improvement in the knowledge of the fundamentals of any subject is almost always the spring-board for further advance in practice. Straight mineral oils only will be considered, and EP oils will not be mentioned at all.

First it will be recalled that kinematically gears can be replaced by rotating disks of the correct diameter and speeds. This idealization greatly simplifies investigation of gear behaviour, both theoretically and experimentally. It must be realized that any idealization, if carried too far, leads one astray. In disks it must be remembered that no account is taken of the effect of tip relief. Tip relief is, of course, extremely important (especially in marine gears), but once the instantaneous point of contact has moved away from the point of engagement, gears should be simulated exactly by disks.

It will then be assumed that disks and gears can be treated interchangeably.

This subject will be treated by listing the main features of gear lubrication, some of which have been studied in a new disk tester put into operation at Pametrada. Then the classical theory will be given, and the conclusions that can be drawn from it will be compared with experiment. Modifications of the classical theory will then be discussed.

Experimental

A new variable slide/roll test machine has just been put into operation. This has been engineered by Messrs British Timken Ltd. (Mr J. Mills, designer). It will be fully described in a later paper, but briefly is as follows :

Basically it consists of a 2-inch disk directly driven by a trunnion-mounted motor. This disk drives a flat plate $4\frac{1}{4}$-inches diameter which can be swung in a circle with the point of contact as centre, *see* Figs 1, 2, and 3. The load is applied through a lever and push-rod system acting behind the plate at the contact point. The frictional drag of the disks can be measured by the torque at the motor housing. The amount of sliding can be varied by altering the angle subtended by the centre of the plate to the contact point and the disk-motor axis. If the angle is zero there is no relative motion of the disk and plate; both move at the same peripheral speed. If the angle is 90° the plate is stationary. The speed of the disk can be varied between 3 and 3000 r.p.m. by a thyratron controlled D.C. motor.

* Pametrada Research Station.

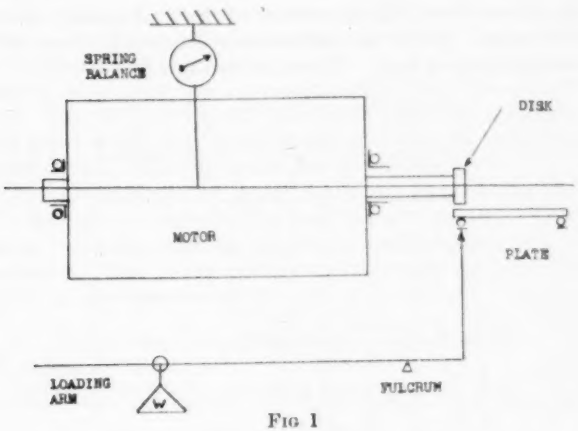

FIG 1

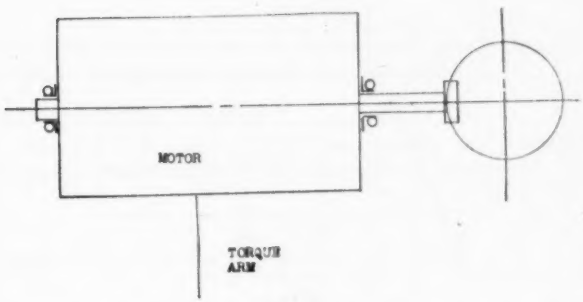

FIG 2

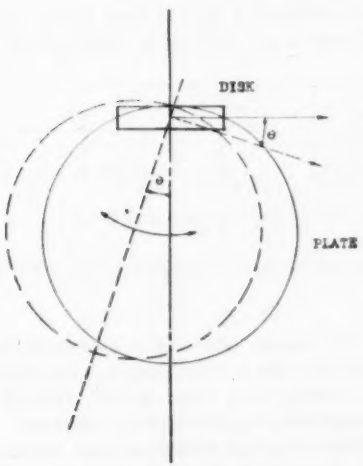

FIG 3

The experiments have shown several of the well-known characteristics of gear lubrication. First, the coefficient of friction is about 0·03 to 0·05, and it is independent of load. This is shown in Fig 4.

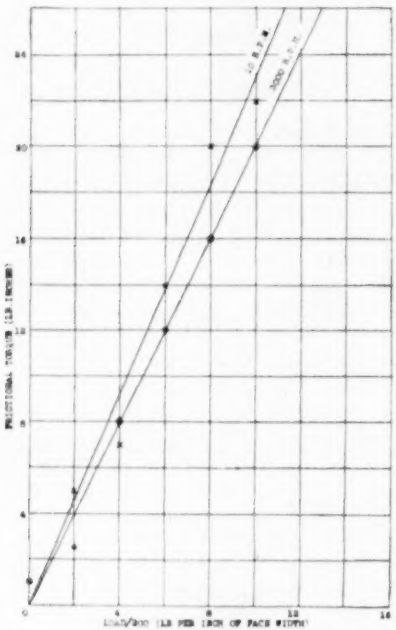

FIG 4

The second point is that the friction usually drops with increasing speed (Fig 5). The interesting part is that at the higher speeds the suddenly scuff. There is no noticeable rising in the friction curve *before*

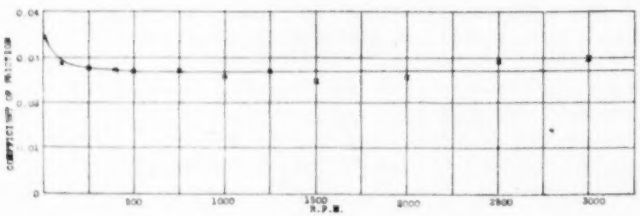

FIG 5

they scuff, when, of course, the curve increases sharply. This would lead one to conclude that the oil film *suddenly* breaks down. It is interesting to notice that scuffing and wear marks, though almost the same to the eye, are quite different under the microscope. A 10 × macrograph of a scuffed disk shows wear as scratches and scuffing as pot holes. The photographs are similar to some published by Borsoff *et al.*[1]

The scuffing of disks (or gears) seems to follow a law of the type : speed^n × load = constant. This has been established by various authors, *i.e.*, Almen,[2] Hofer,[3] and Mansion.[4]

The scuffing load increases approximately with the square root of viscosity.[4, 5] The results of Borsoff can also be plotted to fit this law.

A further point which will be discussed later is that much larger loads can be carried by an oil film than the theory predicts.

Classical theory predicts that disks rotating in pure sliding should carry zero load. Actually a considerable load can be carried. This fact has been reported [6] recently and the necessary modifications of the classical theory discussed. This paper also gives references for the derivation of the equations in the next section.

Theory

I. *Basic Equation*

The basic equation upon which all theory is founded is the expression :—

$$\frac{dp}{dx} = \frac{\delta}{\delta y}\left\{\eta \frac{\delta v}{\delta y}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I)}$$

This neglects the pressure gradient in the axial direction, which, for the conditions under which disks or gears run, is justified.

II. *Rigid Disks, Constant Viscosity*

If it is assumed that the oil is constant in viscosity and pressure throughout the *thickness* of the film, then the equation (I) becomes :—

$$\frac{dp}{dx} = \eta \frac{\delta^2 v}{\delta y^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(II)}$$

which integrates to :—

$$\frac{dp}{dx} = 6(V_1 + V_2)\eta \frac{h - \bar{h}}{h^3} \quad . \quad . \quad . \quad . \quad . \quad \text{(III)}$$

If it is assumed the disks are rigid cylinders, the following expression is obtained for the film thickness :—

$$h = h_0 + R(1 - \cos\theta)$$

where

$$R = \frac{R_1 R_2}{R_1 + R_2}$$

$$\approx h_0\left(1 + \frac{x^2}{2Rh_0}\right)$$

for small values of θ . . . . . (IV)

Combining (IV) with (III), integrating and putting $\tan\psi = \frac{x}{\sqrt{2Rh_0}}$ the following is obtained :

$$p = 3(V_1 + V_2)\eta\sqrt{\frac{2R}{h_0^3}}\left[\left(1 + \frac{3}{4}\sec^2\bar{\psi}\right)\left(\frac{\pi}{2} + \psi + \sin\psi\cos\psi\right) - \frac{1}{2}\sec^2\bar{\psi}\sin\psi\cos^3\psi\right]. \quad \text{(V)}$$

where $\bar{\psi}$ corresponds to $\bar{h}$

with the conditions $p = 0$ at $x = a = +\infty$

$p = dp/dx = 0$ at $x = -b$

The maximum pressure occurs at $x = +b$

$$p_{max} = 0{\cdot}76\,\eta(V_1 + V_2)\sqrt{\frac{R}{h_0^3}} \quad . \quad . \quad . \quad . \quad \text{(VI)}$$

The load carried $W = \int_a^{-b} p\,dx$

$$= 2{\cdot}445\,\eta(V_1 + V_2)\frac{R}{h_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(VII)}$$

The coefficient of friction referred to disk 1 or 2 becomes

$$\mu_{1,2} = 1{\cdot}470\sqrt{\frac{\eta(V_1 + V_2)}{W}}\left(1 \pm 1{\cdot}238\,\frac{V_1 - V_2}{V_1 + V_2}\right) \quad . \quad \text{(VIII)}$$

Combining with (VII)

$$\mu_{1,2} = 0{\cdot}94\sqrt{\frac{h_0}{R}}\left(1 \pm 1{\cdot}238\,\frac{V_1 - V_2}{V_1 + V_2}\right) \quad . \quad . \quad . \quad \text{(IX)}$$

The energy dissipated in the oil film is obtained from the dissipation factor per unit volume :

$$\eta\left\{\frac{\delta v}{\delta y}\right\}^2$$

Thence, integrating over the entire oil film,

$$H = \int_{-\infty}^{\infty}\int_0^h \eta\left\{\frac{\delta v}{\delta y}\right\}^2$$

$$= 2{\cdot}30\eta\sqrt{\frac{R}{h_0}}\{(V_1 + V_2)^2 + 1{\cdot}93(V_1 - V_2)^2\} \quad . \quad . \quad \text{(X)}$$

or incorporating equation (VII)

$$= 0{\cdot}94\sqrt{\frac{h_0}{R}}(V_1 + V_2)\,W\left\{1 + 1{\cdot}93\left[\frac{V_1 - V_2}{V_1 + V_2}\right]^2\right\} \quad . \quad . \quad \text{(XI)}$$

where it is assumed that the viscosity has a constant value from $x = a$ to $x = -b$ and from $x = -b$ to $x = -\infty$. This is taken by analogy with journal bearings.

With the aid of these equations it is possible to go a reasonable distance in explaining the phenomena of gear lubrication.

First, it is seen that the load carried is proportional to

$$\frac{\text{viscosity} \times \text{speed} \times \text{radius}}{\text{film thickness}} \text{ (equation (VII))}$$

Thus thick oil and increasing speed and large radii of curvature and finer finish are all factors leading to a rapid oil-film formation. Thus in a system with increasing speed the friction will drop as the lubrication changes from boundary to full fluid.

As the speed increases the heat generated also increases (equation (XI)), and hence the oil thins and the oil pressure drops. When this is less

than a critical value the surfaces weld together. It has been found experimentally that this can happen without much warning.

It would also be expected that under given conditions there is a maximum load carried at a speed such that the oil film has built up to a satisfactory value at which the heat generation is not so high as to cause excessive thinning of the oil.

Equation (XI) also shows that a relationship of the form (speed × load) gives a breakdown criterion for the maximum amount of heat that can be removed from the system, assuming the oil film is constant. This could come about if the film thickness is determined by the surface roughness; *i.e.*, if the disks ride on the peaks of the roughnesses.

In this connexion it should be pointed out that a criterion having the form (speed$^{\frac{1}{2}}$ × load) is also obtained if it is considered that the flash temperatures of the disks are not to exceed a certain value. This will be discussed later.

This simple theory does not, however, explain why the coefficient of friction is independent of load, as it gives $\mu \propto 1/\sqrt{W}$, unless h_0 is kept constant (equation (IX)).

Secondly, the load carried is much greater than theory predicts. The minimum friction is attained at 750 r.p.m. in the experiment with a load of 1000 lb/inch. Theory gives a film thickness of $5\frac{1}{2} \times 10^{-6}$ inches. The disks each had an initial roughness of 25 micro-inches maximum.

III. *Rigid Disks. Viscosity Varies with Pressure*

The variation with pressure is the simplest with which to deal. There has been considerable work on the subject of the variation of viscosity with pressure and a law of the form $\eta_p = \eta_0 e^{\phi p}$ agrees well with observed results. The coefficient drops with increasing temperature.

Equation (II) becomes :—

$$\frac{dp}{dx} = \eta_0 e^{\phi p} \cdot \frac{\delta^2 v}{\delta y^2} \text{ or } \frac{dp}{dx} e^{-\phi p} = \eta_0 \frac{\delta^2 v}{\delta y^2}$$

then the R.H.S. is unaltered.

The R.H.S. of equation (V) will be the same, the L.H.S. will be $\dfrac{1 - e^{-\phi p}}{\phi}$ instead of p.

Now p_{max} will be infinite when $\phi \times 0{\cdot}76\, \eta_0 (V_1 + V_2) \sqrt{\dfrac{R}{h_0{}^3}}$ reaches unity.

The total load W is again $\int_a^{-b} p dx$, and when $p_{max} \to \infty$ W is actually still finite, though about two and a half times its original value.[8, 9] If the speed is increased the pressure cannot rise (as it is already infinite), but the peak, instead of being infinite over an infinitely small area, becomes infinite over a finite area. Hence the disks become oil borne *whatever the load.*

This explains the observed fact that the friction speed curve is independent of load, that is the drop from mixed friction to full fluid friction is independent of load and dependent only on speed. This question was discussed in a recent paper.[10]

The obvious criticism of this work, which was realized in the paper cited above, is, of course, that an infinite pressure will produce all sorts of other phenomena neglected in this simple treatment.

First, an infinite pressure will cause infinite viscosity, and at that point infinite heat generation, which will reduce η_0 and ϕ. Further, it is assumed the disks are rigid, but, of course, these will deform and alter the shape of the oil film, which will change the pressure curve.

This theory is clearly dependent on the solution of the equations for the deformation of disks under oil pressure. No further progress can usefully be made until these equations are solved.

IV. *Variation of Temperature*

Next arises the question of temperature variation, both in the direction of motion (the x direction) and across the film (the y direction). These are very difficult to take into account.

The film temperature depends on the surface temperatures of the disks. This latter problem has been treated first by Blok [11] and then Jaeger.[12] The mathematical difficulties are such that only an idealized case can be considered. It is assumed that a *band* source moves along the surface of the disks, the strength of the source is given by half the frictional drag $\times$ sliding speed, it being assumed half the heat goes into each disk. The difficulties of the partition of heat between the disks has been discussed by Jaeger, and a method that appears capable of solving them has recently been perfected by Wassermann.[13] It is hoped to have some computed results on this problem shortly.

Blok's method gives an interesting criterion for failure. He assumes a given oil can sustain a given temperature flash, about 500° C, and from this calculated the load and speed necessary to achieve it. The resulting criterion for any set of gears is of the form

$$\text{load}^{\frac{1}{2}} \times \text{speed}^{\frac{1}{2}} = \text{constant}$$

$$\text{or of course, load} \times \text{speed}^{\frac{1}{3}} = \text{constant.}$$

It is difficult to follow the theoretical basis for this failure criterion, unless it is assumed that melting or welding of the metal occurs at the given breakdown temperature flash, independent of the oil.

It may be that there is some protective oxide or other coating on the surface that a given temperature and pressure breaks down and causes the surfaces to come into intimate contact, with subsequent welding. This is an extremely important subject on which little work has been published.

The variation of temperature across the oil film is also extremely difficult to deal with mathematically. A recent paper [6] has shown that if the temperature obeys a known law, then an oil film can be formed between disks whose surfaces are moving in equal and opposite directions.

Briefly this comes about as follows

$$\text{Equation (i) states } \frac{dp}{dx} = \frac{\delta}{\delta y}\left\{\eta \frac{\delta v}{\delta y}\right\} = \eta \frac{\delta^2 v}{\delta y^2} + \frac{\delta \eta}{\delta y} \cdot \frac{\delta v}{\delta y}$$

It is usual to neglect the last term, and in so doing the load-carrying capacity comes out to be directly proportional to the rolling velocity

$(V_1 + V_2)$ (equation (VII)). If, however, this last term is included, the sliding velocity $(V_1 - V_2)$ also gives a load-carrying component.

Hence even if $(V_1 + V_2)$ is zero the sliding velocity $(V_1 - V_2)$ will carry a load.

V. *Deformation of Disk Surfaces*

The greatest unknown in gear-lubrication theory at the moment is the effect of oil pressure on the shape of the surfaces. Work is in progress on the problem, and so far it looks as if, under heavy deformation, the disks flatten and the pressure curve approximates to the Hertzian elastic pressure.[14] If this is so there is an interesting alternative explanation to the problem of disks getting oil-borne independent of load.

Assume the disks become flat following the Hertzian pressure distribution and that they obey the laws of slider bearings.

If this is so, then the oil film thickness $h_0 \propto L\sqrt{\frac{\eta V}{W}}$, but L the width of the Hertzian flat is proportional to $\sqrt{W}$. Hence $h_0 \propto \sqrt{W}\sqrt{\frac{\eta V}{W}} \propto \sqrt{\eta V}$.

This is an interesting speculation, and further work will show if it is true or not.

VI. *Other Effects*

There are various other possible effects in gear lubrication. The roughness of the surfaces is an unknown. Work on Michell pads shows that the same general lubrication laws hold, but the constants for film thickness etc. are altered.

A mathematical study of the effect of grooves in the direction of motion has been published by Michell.[15] This is being extended at Pametrada, both to cover the region where the amplitude of the grooves is of the same order as the roughnesses and also where the grooves run at right angles to the direction of motion.

There is, however, the suggestion [16] that opposing roughnesses can trap oil between them and thus support a load. This mechanism might occur when there is continuous contact between the surfaces, but it is hard to see how it would alter the normal Reynolds' equations when there is a continuous oil film supporting the load.

Finally, to be mentioned is the theory of Heidebrock,[17] recently summarized in the *Z.V.D.I.*, that there is a layer of oil that attaches itself to the oil flanks. This oil layer is several hundred Ångstroms thick. The difficulty that surface chemists experience is to understand the mechanism of such long-range orientation forces. The author often wonders if a lot of these orientation effects are not in some way bound up with the surface roughnesses which are of the order (10^{-5} to 10^{-6} inch) of the oriented layer as postulated.

Conclusion

In conclusion, it is clear that hydrodynamic theory does explain quite a number of features of gear lubrication. The quantitative side of the picture is, however, not so satisfactory.

Further progress is limited by the solution of the problems of the deformation of the gear surfaces and the temperature distribution in the oil and the rubbing flanks.

The work has gone far enough, however, to show that normal hydrodynamics will give the answers to quite a number of problems in gear lubrication.

Acknowledgment

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List of Symbols

a *and* b	extent of the oil film.
h	film thickness at any point.
$\overline{h}$	film thickness when $dp/dx = 0$.
h_0	minimum film thickness.
H	energy dissipated by oil.
L	length of Hertzian flat.
p	oil pressure at any point.
$R_{1,2}$	radius of disks 1, 2.
v	velocity of a particle of oil.
$V_{1,2}$	velocity of surfaces of disks 1, 2.
W	load per axial length carried by oil film.
x	axis in direction of motion.
y	axis in direction of film thickness.
η	viscosity at any point.
η_0	viscosity at atmospheric pressure.
η_p	viscosity at pressure p.
θ	angle subtended to the centre by any point on the surface and centre of second disk.
ϕ	pressure–viscosity coefficient.
ψ	substitution angle in equation (v).
$\mu_{1,2}$	coefficient of friction referred to disks 1, 2.

GEAR-TOOTH WEAR

By G. I. Finch and R. T. Spurr *

The engagement of the load-bearing areas in the gear assembly may be regarded as compounded of rolling and sliding motions. It is characteristic of the gear in general that the load intensities are high, while, with certain exceptions (*e.g.*, the hypoid gear), the rates of sliding are low. Further, as the region of engagement changes, these quantities vary in magnitude. In the practical gear assembly this simple picture is complicated by housing and shaft flexure, inaccuracies in journal fitting, tooth profile and surface finish, load transfer from tooth to tooth, and elastic deformation and hysteresis in elastic recovery of the contacting surfaces. These complications may give rise to variations, not only in the load intensity but also in the relative sliding speeds, even to the extent of causing momentary reversals in the direction of sliding.

In considering the mechanism of gear-tooth wear, therefore, it is necessary to envisage a high intensity of loading (*a*) without shear between the contacting surfaces, (*b*) with a low variable shear, and (*c*) with a high, generally constant rate of sliding.

(*a*) It is well known that, when two clean metallic surfaces are brought into contact, cohesion occurs across the interface and causes them to seize together. In practice, however, bearing surfaces are covered with thin films of oxidation products, adsorbed gases and vapours, and lubricant molecules, and it is between these films that contact occurs. The cohesive forces between such layers are small compared with the forces between the atoms in the bulk of the metal; hence, on parting the surfaces, separation occurs relatively easily within these non-metallic layers. But if the loading intensity is sufficiently high to cause such deformation of the underlying metal that the surface layers become stretched and ruptured so that metal is exposed, intermetallic contact and strong cohesion may ensue. It is evident, therefore, that in this case resistance to seizure is not primarily determined by the properties of the surface layers, but is related to the ability of the underlying metal to resist deformation and hence to support these layers. Consequently, surfaces which are pressed together without shear to a high loading intensity should consist of metals of high rigidities. A similar conclusion may be drawn for surfaces in rolling contact, as occurs in part of the contact cycle of the gear-tooth. Some measure of shear, which can be intermittent, may, however, occur between the surfaces, owing to elastic deformation and hysteresis in elastic recovery. In the pitch-line region of the gear-tooth, where the rolling is between surfaces of similar curvature, elastic deformation should not result in shear, provided the surfaces have the same rigidity. When a critical load intensity is exceeded the surfaces may rumple, or suffer fatigue failure after repeated application of the load.

(*b*) The severity of conditions is accentuated when the purely rolling mechanism of progressive load transfer from one region to another of the mating surfaces is accompanied by sliding shear. It now becomes possible

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for any boundary lubricant film to be sheared or rubbed away to expose the oxide layers. These in turn may become disrupted and give rise to compacted particles of debris which become foci of excessive loading intensity leading to plastic deformation of the underlying metal and possible local seizure. Once formed, such a weld tends to increase in extent as a result of the sliding motion. A weld will not, in general, shear in the plane of sliding; for the material at the junction is work-hardened, and the distribution of stress about the junction is generally such that failure will occur within the bulk of one surface, so that a fragment of metal will be torn out of one surface and left adhering to the other. This fragment will then score and tend to seize again to the surface sliding past it, and so the damage may become progressively worse.

(*c*) When the sliding speeds are high as, for example, in the hypoid gear, the frictional heat generated between the sliding surfaces becomes important. Owing to surface imperfections the load is carried at isolated contact areas instead of being uniformly distributed over the whole mating area. Whilst the total heat energy developed is proportional to the product of the load and the velocity of sliding, the temperature rise at the areas of contact is primarily determined by the velocity. The temperature reached at a surface irregularity may be so high that the metal begins to soften and flow, but provided the surface films on both surfaces are not simultaneously displaced, the irregularity will tend to be smoothed away and the surface will develop a polish layer. The opposing surface, along which the irregularity rubs in a long contact path, does not reach such high temperatures. With sufficiently severe sliding conditions, however, the temperature may rise to such values that the rigidities of both surfaces are impaired and intermetallic contact and seizure result through the disruption of the surface film. The function of the extreme-pressure additive is to form a surface compound layer whose melting point lies well below that at which the rigidity of the supporting metal becomes impaired.

CRITERIA GOVERNING SCUFFING FAILURE

By F. T. BARWELL * and A. A. MILNE *

SUMMARY

Experiments in reciprocating motion do not provide a clear basis for discrimination between WN and $\mu W\sqrt{N}$ as criteria for scuffing failure, but where motion is continuous $\mu W\sqrt{N}$ appears to apply. In neither case does $W\sqrt{N}$ approach constancy. (Here W = load, N = speed, μ = coefficient of friction.) The existence of hydrodynamic pressures has been demonstrated experimentally, and scuffing load has been shown to depend on the method used to finish the surfaces.

Examination of scuffed surfaces shows that material transferred from one surface is deposited in several layers on the opposing surface, which is itself deformed plastically. Micro-hardness determinations of the scuffed material show that it is very hard and the very fine structure is indicative of rapid cooling.

It is concluded that gears should be designed to establish fluid-film lubrication conditions, but that, where this is not possible, attention to surface finish and material may enable higher loads to be carried.

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Notation

W = Total load on mechanism to cause scuffing.
N = Speed of mechanism.
μ = Coefficient of friction.
p = Load per unit width of tooth.
P = Compressive Hertzian stress.
V = Sliding velocity.
T = Length of action from pitch point to end of contact. (cf. Ref. 1.)
t = Temperature flash °C.
$a = \sqrt{\lambda C}$.
λ = Heat conductivity.
γ = Density.
C = Specific heat.
R = Half width of contact.

Introduction

Although there has been considerable laboratory experimentation on gear lubrication, the information that has thus been obtained has so far proved to be of little use in the design of gears. The practice appears to be that the draughtsman designs gears on the basis of certain empirical relations tabulated in various handbooks and specifications, and then requests a lubricant from an oil supplier to suit. From experience with similar gears in operation, the oil technologist is usually able to provide a satisfactory lubricant. When a particular design happens to produce especially severe operating conditions, as evidenced by frequent service failures, a stimulus is produced for the development of new lubricants. Laboratory tests simulating service conditions discriminate approximately between possibilities, but trial under service conditions is the final arbiter. The main feature of this procedure is that the lubricant is developed to suit the existing design—present knowledge is insufficient for design to suit available lubricants.

Empirical relations governing the rating of gear designs have been accumulated over a period of time. These are based mainly on the strength aspects of the gear material. Comparatively recently, however, Almen [1] has produced a "scoring factor" that seems to make due allowance for susceptibility to scuffing. Quoting a large number of practical cases, he shows that for gears of similar design a criterion based on PV, where P is the compressive Hertzian stress and V is the tooth sliding velocity, is reasonably satisfactory. For gears of different design another factor T, the length of action measured from the pitch point to the end of action, is needed. This modified PVT factor seems to hold well for gears of similar material when lubricated with a mineral oil. There is as yet insufficient evidence for the formulation of similar criteria for extreme-pressure lubricants or for wider applicability.

Manhajm and Mansion [2] show curves (for "oil 19" and "oil 7") tested on the IAE machine, which support a WN criterion. Moreover, they reveal no general dependence on viscosity; it is regarded as a minor factor compared with the differences in oils. Lane and Hughes,[3] however, produce results indicating that $N^{2/3}\ W$ = constant, and go on to produce an expression relating failing load with viscosity as well as speed. Lane,[4] using the two-ball machine, arrives at the conclusion that the performance of a lubricant is determined by the attainment of a characteristic tem-

perature. Theoretical justification for this result is available from consideration of the classical work of Blok [5] on calculation of flash temperature.

The dependence on viscosity shown by Lane, and in particular oils by Manzion, may be due to differences in composition of the oil which are revealed by differences in viscosity at low temperatures, but which lead to other differences in behaviour at high temperatures and pressures. In this connexion the work of Clarke, Woods, and White [6] using crossed cylinders is particularly appropriate, in so far as their work leads to the conclusion that the oils provide protection due to the enormous increase in viscosity at high pressures. They attempt to show that some oils become solid and that the temperatures reached are insufficiently high to destroy the effect.

Blok gives an expression for the magnitude of the momentary temperature flash for two different cylindrical surfaces in a combination of rolling and sliding motion in the form—

$$t \propto \frac{\mu p(V_1 - V_2)}{(a_1\sqrt{V_1} + a_2\sqrt{V_2})\sqrt{R}}$$

In his experimental confirmation Lane ignores any variation in coefficient of friction. It is possible that in his particular experiments the coefficient remained sensibly constant, but if the criterion is to have general applicability, then due allowance must be made for variations in friction that undoubtedly occur in thin-film lubrication.

Experimental

Because scuffing is usually experienced at the tips of gear-teeth, where sliding is at a maximum, and seldom in the vicinity of the pitch line, where rolling predominates, the sliding component may be regarded as being mainly responsible for surface damage. The rolling component may be expected to operate in favour of setting up hydrodynamic conditions. The authors' experiments were therefore made under conditions of pure sliding, using flat-ended rectangular pegs running against flat surfaces. Two machines were used, one in which the flat surface was reciprocated [7] and the other in which it took the form of a disc rotated at constant rate. Results are given in Tables I and II.

Table I

Reciprocating Motion

Materials = Tool steel on mild steel.
Lubricant = Mobiloil No. 57.
Stroke = 4·5 inches.
Contact area = 0·024 sq. in.

Speed r.p.m.	Scuffing load, lb	Coefficient of friction before scuffing	Scuffing criteria		
			WN	$W\sqrt{N}$	$\mu W\sqrt{N}$
250	>200	—	>50,000	>3,100	—
500	115	0·04	57,500	2,600	104
1,000	45	0·11	45,000	1,400	154
1,500	30	0·16	45,000	1,100	176

TABLE II

Circular Motion

Materials = Mild steel on mild steel.
Lubricant = Intava 620.
Diameter = 3 inches.
Contact area = 0·037 sq. in.

Speed r.p.m.	Scuffing load, lb	Coefficient of friction before scuffing	Scuffing criteria		
			WN	$W\sqrt{N}$	$\mu W\sqrt{N}$
2,000	365	0·048	730,000	16,300	783
3,000	405	0·038	1,220,000	22,200	843

It was noted that there was a considerable drop in coefficient of friction as the load was increased; this continued until scuffing occurred in reciprocating motion, but was reversed before scuffing took place in circular motion. The values quoted in the table were those measured immediately prior to scuffing.

Applied to these experiments the temperature criterion reduces to

$$t \propto \mu p\sqrt{V}$$

because $V_2 = 0$, R and a_1 are constant.

The bulk temperatures at scuffing were approximately the same, and were small compared with the estimated temperature flash. The criterion for these experiments is therefore proportional to $\mu W\sqrt{N}$, and is shown in Table II to give a fair correlation with the results of the circular tests. The alternative criteria, WN and $W\sqrt{N}$, do not give such a good agreement. In the case of reciprocating motion the WN criteria gives closer correspondence than the $\mu W\sqrt{N}$, although it is again possible to eliminate the $W\sqrt{N}$ relationship.

The variation in coefficient of friction makes it apparent that conditions of lubrication are not purely boundary, but are in the mixed or thin-film region. This is a region about which little is known, but it is believed that a substantial portion of the load may be carried by fluid pressure. In order to determine whether such fluid pressures were entering into these scuffing experiments a small hole (0·03 inch) was drilled in one of the contact areas and connected to a pressure gauge. Substantial pressures were recorded on this gauge, building up slowly as the load was increased. Typical curves showing the variation of pressure and friction with increasing load are shown in Fig 1. It will be seen that the pressure reaches a maximum at a load rather higher than that giving the minimum friction and thereafter drops slowly. The ratio of this pressure to the mean pressure on the surfaces is about 2 at very low loads and steadily falls. At the point at which scuffing occurs there is still a considerable pressure (ratio about 0·4) present.

If thin-film conditions are operating during the critical scuffing range, the degree of surface finish will play an important part in determining the scuffing load. This is demonstrated in Table III, where the experiments

were similar except in the mode of finishing the moving contact surface. It will be noted that alterations in finish may effect a two-fold change in scuffing load, both the magnitude and relative direction of the rugosities being important. It appears that a slight irregularity across the direction of motion, but not too much, is the most suitable condition. Whether

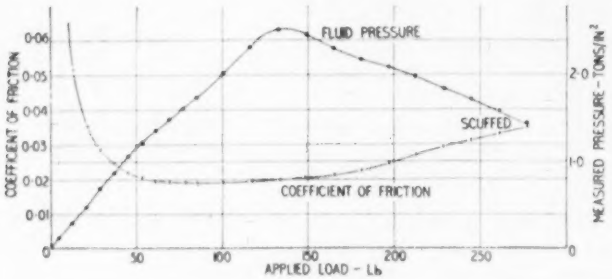

FIG 1

VARIATION OF FLUID PRESSURE AND COEFFICIENT OF FRICTION WITH LOAD IN CIRCULAR MOTION

TABLE III

Effect of Surface Finish

Apparatus = Reciprocating machine.
Materials = Hardened steel on hardened steel.
Lubricant = Intava 620.
Stroke = 4·5 inches.
Contact area = 0·025 sq. in.
Bulk temperature = 100° C.
Speed = 1000 r.p.m.

Surface finish	Scuffing load, lb
No. 2 emery cloth—across	85
No. 1½ „ „ —across	75
No. 1F „ „ —across	65
Ground—along	50
No. 1F „ „ —along	60
No. 1½ „ „ —along	35
No. 2 „ „ —along	35
Polished to 4/0 emery, random	90
No. 2 emery cloth, random	40

these irregularities are beneficial by providing oil reservoirs, promoting fluid pressures, or by shortening the momentary contacts, is as yet unknown, and merits further consideration.

SURFACE DAMAGE

Macro Effects

When scuffing occurs fusion and welding of the surfaces takes place, and a certain amount of material is usually transferred from one surface to the other. A typical appearance of scuffing is shown in Fig 2. It will be noted that there are a number of discrete blobs of material deposited.

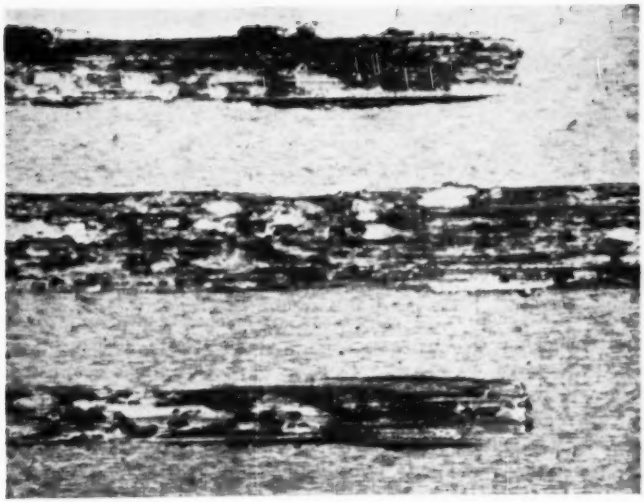

FIG 2

A SCUFFED SURFACE (×6)

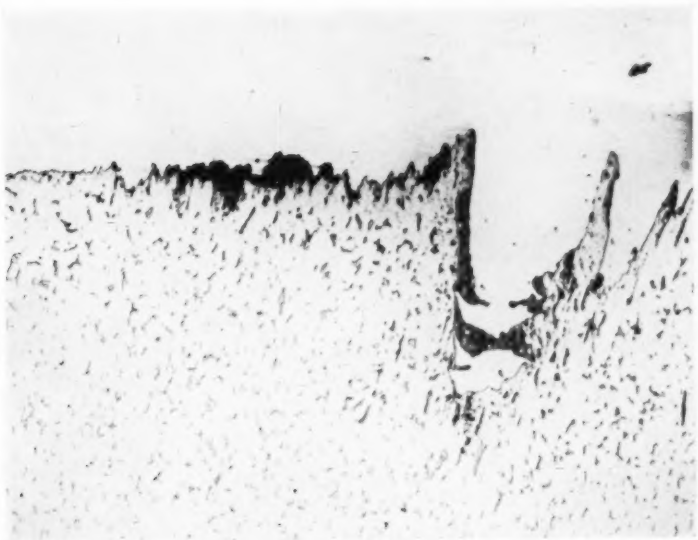

FIG 3

TAPER SECTION THROUGH A SCUFFED SURFACE (H × 50, V × 500)

[To face p. 628.

When speeds are low and the loads high, these blobs are large but few in number; when the speeds are high and the loads low there are more blobs, but each is comparatively small. If the process is allowed to continue, these blobs eventually spread to cover the whole contact area.

At very high sliding speeds this phenomenon of transferred material may be superseded by a rather different one. Instead of most of the removed material being deposited on the other surface, material is removed from both surfaces in small particles, and is carried away as debris in the oil. "Scuffing," in the conventional meaning of the word, does not occur at all.

Micro Effects

Examination of the surfaces which have been run at loads just below that causing scuffing reveals little evidence of change. The surfaces sometimes appear polished, with the familiar glazed appearance of well-run-in bearing surfaces. No changes have yet been detected that enable prediction of impending failure to be made. After scuffing, however, the surfaces are considerably altered. Sakmann,[8] Clayton and Jenkins [9] have shown that chemical and metallurgical changes are both profound, and indicate that very high temperatures followed by rapid cooling have been attained at the points of failure. The resulting damage may extend well into the body of the surface. Fig 3 shows a typical taper section through a scuffed area. The transferred material is deposited in several layers and extends well into the bulk material, with considerable plastic deformation all around. In this particular specimen there has also been some tearing of the adjacent surface. Micro-hardness determinations of this scuffed material show that it is very hard, and taken in conjunction with the very fine structure show that the process of fusion and solidification has taken place with great rapidity.

Related Surface Phenomena

General

In considering the form of surface failure known as scuffing it is immediately apparent that many other phenomena reported in the literature have some relevance. A complete description of all these factors would be out of place in this brief survey, but mention must be made of some of the most important. These are all concerned with the interaction of sliding surfaces at the points of intimate contact.

Temperature Flash

As scuffing is characterized by fusion of the metal surface, it is obvious that extremely high temperatures must be reached momentarily at the points of contact. This aspect of scuffing has received closest attention theoretically from Blok [5] and Lane,[4] who have endeavoured to show the existence of a critical temperature which, when reached, leads to failure. Bowden [10] has shown the existence of high local temperatures even at very low speeds of sliding, while the calculations of Jaeger [11] have also shown that very high temperatures are probable.

Growth of Junctions

It has been observed, by Parker [12] and also by Finch,[13] that the damage caused by sliding may vary with the distance of sliding. With some combinations of materials the area of contact tends to increase as sliding progresses, with others it tends to decrease. It may well be that this phenomenon is closely related to that of recovery in scuffing experiments. If the surfaces are susceptible to small amounts of damage, then gross scuffing will be initiated by slight imperfections in the surfaces. If the surfaces have some measure of self-protection from small disturbances, then scuffing may be delayed.

Equations of Wear

Burwell's [14] recent work on wear may well have an important bearing on critical scuffing conditions. He showed that the equations developed from theoretical considerations for the wear of surfaces were reasonably valid up to a critical value of applied load, but that above this load very severe wear took place. The magnitude of the critical load corresponded approximately to one-third of the hardness of the material. He postulated that the sudden increase might be attributable to the interaction, above this load, of the damage or debris from the numerous previously isolated areas of intimate contact.

Although this simple value of H/3 is clearly inappropriate to scuffing loads as determined by experiment, for they vary considerably with speed, the concept of interaction between junctions may well be of importance, especially if coupled with the idea of growth or inhibition of junctions. The bases for these concepts are at present ill-established, and further investigation would be desirable.

Critical Lubricant Temperatures

Ordinary oiliness additives are notoriously ineffective as extreme-pressure lubricants. Their action at normal temperatures is believed to depend on the orientation of their molecules on the metal surface or on the rheological properties of their reaction products. Tabor has shown that besides the well-known disorientation temperature at about 100° C for fatty acids, there exists a higher critical temperature at which lubrication is completely ineffective, and friction approaches that for dry surfaces. It is probable that the crucial region for extreme-pressure lubricants is above this higher critical temperature. If extreme-pressure lubricants are effective owing to their reaction with the surfaces, then it is also possible that their reaction products themselves may have still higher critical temperatures that correspond to those postulated by Blok.

Theory of Failure

On the basis of these considerations we are able to formulate a tentative theory of failure. This may be summarized as follows :

" During sliding of the surfaces there will always (except under full fluid lubrication conditions) be a few points of momentary intimate contact between the rubbing surfaces. At these points the pressures and the

coefficients of friction will be high and considerable heat will be generated. At low loads and speeds this heat is rapidly dissipated in the lubricant and to the surrounding bulk material. At some critical condition of load and speed, depending on the rubbing combination, the increased temperature will lead to a change in area of contact, coefficient of friction, or other factor, sufficient to cause an unstable temperature system rising until the melting point of the material is reached."

It will be noted that this theory does not necessarily include the postulated critical temperature of Lane. Such a critical temperature, however, may well exist, at which the friction rises considerably, and this may possibly coincide with or be related to the higher critical temperatures mentioned by Tabor.

The unelaborated theory, as stated above, is sufficient to account for the observed phenomena of scuffing. There is no reason, however, why scuffing should be regarded as an isolated phenomenon. The basic cause, that of intimate metallic contact, is common to all conditions of sliding under mixed or boundary lubrication. Under any conditions wear may be regarded as being due to the surface stresses exceeding a critical value : it merely happens that in scuffing the material is molten when this occurs.

Prevention of Failure

Gear Design

The most efficient way of preventing failure by scuffing is by design of the gears to establish full fluid-film lubrication. Numerous calculations based on hydrodynamic theory have been made by Martin,[15] Cameron,[16] Gatcombe,[17] and others to determine the load-carrying capacity of rolling and sliding surfaces. Early calculations based on simple theory tended to show that the maintenance of fluid films was only possible at very light loads. Recent advances, however, taking into consideration the effect of elastic deformation of the surfaces, viscosity, and density variations of the oil, and also the effect of changing velocities, are leading to the conclusion that it might be possible to establish substantial fluid films under certain conditions. Analysis of these problems is still continuing.

In spite of these predictions of hydrodynamic theory it appears that most gear combinations, especially when approaching failure, are operating under thin-film or mixed-lubrication conditions. The state of knowledge regarding this region is still obscure, although it is attracting increasing attention. Indications are, however, that by suitable choice of factors there is the possibility of generating considerable film pressures on a microscopic scale tending to support the load.

Surface Characteristics

However successful the design may be in fluid-film formation, the problem ultimately resolves into the prevention of seizure under boundary lubrication conditions. If seizure occurs when a certain critical temperature is reached, then protection may be sought either in reducing the heat generation or in raising the critical range; these two factors may not, of course, be independent. The former may be achieved either by reducing

the friction at the points of contact, by a greater heat dissipation, or by reducing the time of momentary contact.

Conventional boundary lubricants have usually been found to have little effect on scuffing characteristics. It is clear that oiliness at room temperatures is irrelevant; but it is by no means improbable that the action of extreme-pressure lubricants is due in part to a capacity for giving low coefficients of friction at high temperatures. Their corrosive character may thus be considered analogous to the chemical attack of fatty acids on metals that is believed to be the cause of their effectiveness at room temperatures. Heat dissipation by improved thermal conductivity may possibly lead to some improvement, but the range of conductivities of materials suitable for gears from strength and fatigue requirements is slight. A deliberate, though slight, roughening of the surfaces may be beneficial as a protection against scuffing. This has normally been attributed to the formation of oil reservoirs in the irregularities; it may well rather be due to the interruption of heating flashes before they produce too high a temperature. Another method might be to control the time in contact by altering the hardness of the surface material.

At the postulated critical temperature, conditions rapidly deteriorate until welding occurs. Improvements may be effected by reducing the affinity of the rubbing surfaces for each other, which may be equivalent, in fact, to a reduction in friction. Two methods of proved value are the use of materials of entirely different composition, such as cast iron and bronze, and the coating of the metal with a layer of phosphate crystals or similar material. Numerous possibilities exist in this line, and the problem resolves itself into a question of determination empirically of the most satisfactory process.

Acknowledgment

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SOME FACTORS AFFECTING GEAR SCUFFING

By H. D. Mansion *

Introduction

During the last fifteen years or so it has been found increasingly desirable to include in gear-oil specifications some measure of their anti-scuffing or welding-inhibiting properties.

The chief factors responsible for this requirement have been :—

(*a*) the general introduction of hypoid gears for automobile back-axle drives;

(*b*) the need to ensure that the lubrication of the propeller reduction gears of piston-type aero-engines with the ordinary engine oil should remain adequate, despite frequent increases in engine output; and

(*c*) the introduction of the gas turbine, in which it has been necessary to lubricate reduction gearing with the very-low-viscosity oils demanded for these machines by other considerations.

Before discussing the anti-scuffing properties of lubricants, it is necessary first to define the meaning of the term. The failure known as scuffing is a rapid form of local welding and tearing apart of minute particles from the opposing surfaces of bodies in sliding contact. On gears it tends to occur on the faces and flanks of the teeth under conditions of heavy surface loading in the regions of maximum sliding velocity. This failure, which is also known as " galling " or " scoring " in America, is the result of local breakdown of the adherent boundary layer of the lubricant, and must be clearly distinguished from pitting, the other form of surface failure which occurs on gear-teeth (usually near the pitch zone) and which is a surface fatigue failure of the gear material caused by excessive surface stress.

To denote the anti-scuffing value of a lubricant, the term " film-strength " has been frequently used, but this is to be deprecated, since a fluid does not possess such a property. The lubrication of highly loaded gears is generally acknowledged to be of boundary type—with very little assistance from any hydrodynamic film—and the scuffing load is a function of several other variables in addition to the properties of the boundary layer.

A number of laboratory test machines have been devised, and are in use, for assessing the anti-scuffing properties of lubricants. In some of these an attempt has been made to reproduce the type of combined sliding and rolling action which occurs between gear-teeth, while in others the action is one of simple sliding, but the rating of lubricants by these machines is not always found in practice to agree with their rating in service as gear lubricants. It would seem that the best method of testing gear lubricants is by lubricating gears with them.

The present paper gives a summary of investigations into some of the factors affecting gear scuffing, which have been carried out over a number of years by the Motor Industry Research Association and its predecessor the

* The Motor Industry Research Association.

Institution of Automobile Engineers—Research Department. These investigations were made in the IAE 3¼-inch gear-lubricant testing machine, in which actual toothed gears are used as test specimens. A description of the machine, of the gears used in it, and of the development of the machine is also included.

Historical

In 1935 the Automobile Research Committee of the Institution of Automobile Engineers decided to undertake research into the load-carrying capacity and lubrication of gears. To start the investigation, a small power-circulating gear-testing machine (designed by H. Blok) was kindly provided by the Delft Laboratories of the Royal Dutch Shell Group.

Later, two larger machines, operating at 3¼-inch centre distance, and capable of transmitting greater tooth loads, were built, and these formed the prototypes of the present IAE 3¼-inch gear-lubricant testing machine. With these early machines, in addition to some work on automobile gear materials, investigations were made into the effect on gear scuffing of various types of lubricants and additives, of chemical surface treatments, and of electrodeposited metallic coatings. The results of these investigations form the main part of this paper.

In 1943–44 the question of suitable additives for the lubricating oil of the Rolls-Royce Merlin engine, as then being produced, assumed considerable importance, since periodic increases in developed horsepower had caused a certain amount of trouble with scuffing of the propeller reduction gears. Some half-dozen oils, including a mineral oil doped with various additives of the mild EP type, and castor oil, were submitted by the Ministry of Aircraft Production for comparative rating by a number of investigators in various lubricant-testing machines, and it was found that the comparative ratings of the anti-scuffing properties of these oils, as given by the IAE 3¼-inch gear-lubricant testing machine, correlated more closely with those of bench tests in Merlin engines than the ratings given by any of the other lubricant-testing machines used.

The MAP therefore decided that a number of 3¼-inch gear-lubricant testing machines, basically resembling the two prototypes, but of improved design, should be built and distributed amongst the interested parties. A contract for this work was placed with Messrs W. E. Sykes Ltd. of Staines, and was carried out under the supervision of the IAE.

Much work has been carried out by the various operators of these machines in the U.K., and whilst they have not yet been adopted as a standard means of specifying gear oils, they have proved useful tools in the continual search for improved gear-oil additives.

The IAE 3¼-inch Gear-lubricant Testing Machine

The gear-lubricant testing machine is of the power-circulating type in which two pairs of gears are loaded torsionally against one another, the driving motor being required to provide only the frictional losses. The two pairs of gears are housed in separate gearboxes mounted at opposite ends of a bedplate, which in turn is mounted on top of a large cast-iron pedestal of box form with a central partition, see Fig 1.

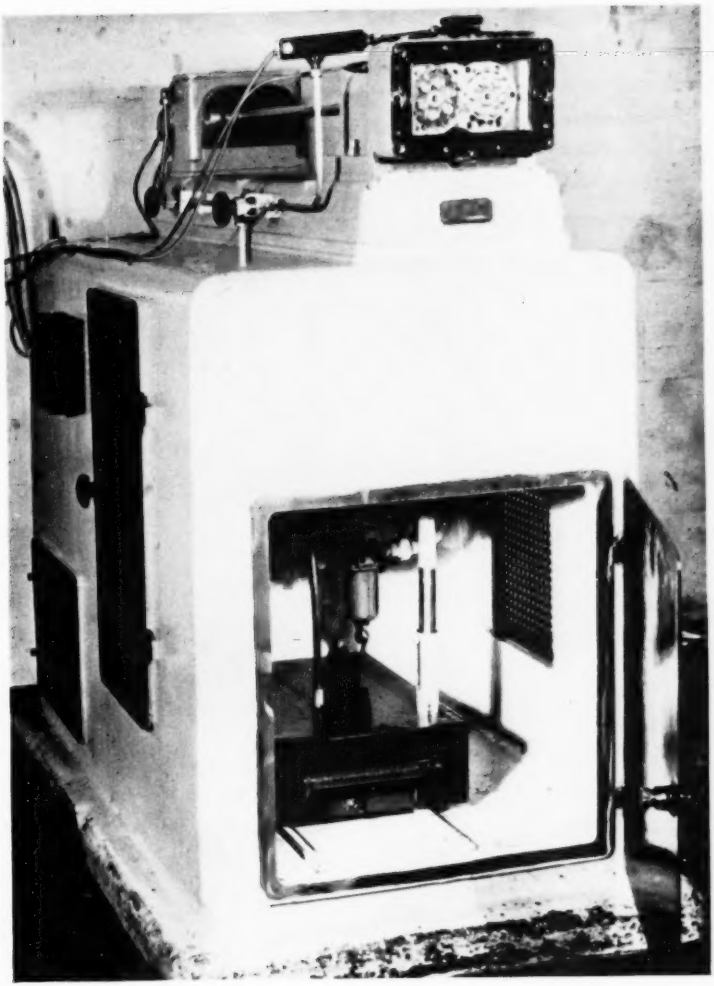

FIG 1

IAE 3¼-INCH GEAR LUBRICANT TESTING MACHINE

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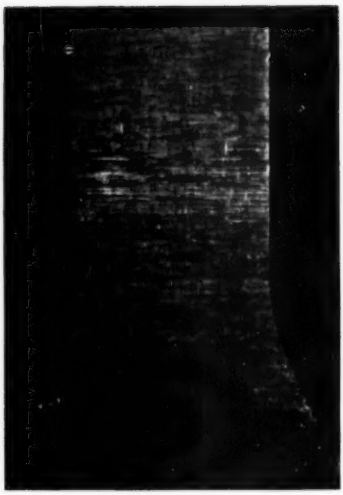

FIG 3

FACE OF TYPICAL TEST GEAR, AS GROUND. × 6

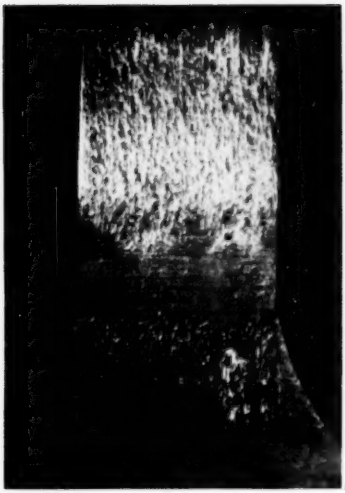

FIG 4

FACE OF TYPICAL SCUFFED GEAR. × 6

The test gears are of coarse-pitch spur type, and are located in a glass-fronted gearbox at the front of the machine. They are spigot-mounted on overhung gearshafts, which are connected by torsionally flexible coupling shafts (through the medium of splined sleeves) to two other gearshafts carrying double-helical power-return (or "slave") gears. These are of similar tooth ratio, but finer pitch, than the test gears, and have greater face width.

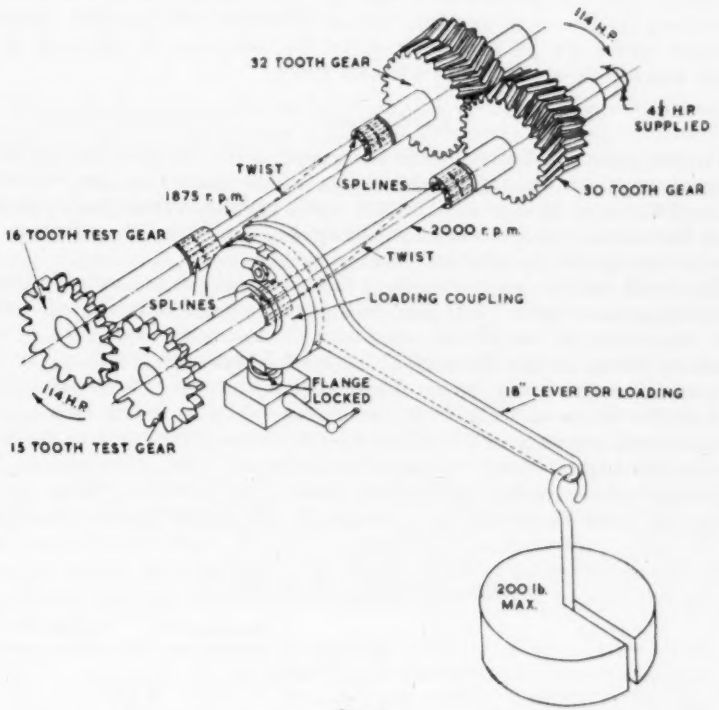

FIG 2

DIAGRAMMATIC ARRANGEMENT OF IAE 3½-INCH GEAR-LUBRICANT TESTING MACHINE, SHOWING METHOD OF LOADING

One of the coupling shafts is divided at one end by a loading coupling, of which one half can be locked against rotation while a torque is applied to the other half by means of a lever and weights, as shown in Fig 2. The torsional deflection, or twist, thus produced in the coupling shafts can be locked up prior to running by bolting the coupling halves together.

The machine is belt-driven, through an extension of one of the power-return gearshafts, by a 4½-h.p. AC motor mounted within the pedestal, though experience has shown that, for running at high speeds not originally envisaged, but found to be necessary for the testing of hypoid lubricants, a larger motor of 10 to 15 h.p., mounted externally, must be used.

The oil system, comprising a tank, gear pump, edge type and magnetic

filters, electric heater, by-pass valve, and the necessary piping, is located within the other compartment of the pedestal, seen open in Fig 1. As designed, the lubrication of both ends of the machine was from one common supply, but it is understood that some users have modified their machines so that each end has its own lubrication system, thus minimizing the amount of test oil required.

The oil temperature is indicated and controlled to within $\pm 1°$ C by a combined dial thermometer and regulator, the bulb of which is located in the oil supply line close to where the oil enters the test gearbox. The oil pressure above the jet supplying oil to the test gears is indicated by a gauge and can be controlled by a needle valve.

Test Gears

The test gears are of fifteen and sixteen tooth, 5 D.P. design with a working pressure angle of approximately 26 deg when running at the extended centre distance of $3\frac{1}{4}$ inches. The face width is $\frac{3}{16}$ inch from the tip to the pitch line, below which it widens out to the $\frac{1}{2}$ inch width of the gear body to give the maximum possible bending strength.

The tooth profiles are ground by a formed-wheel process with a slightly modified involute form. For the earlier tests a tip and root relief of 0·0002 inch was aimed at, but for the later tests the tolerances laid down by the Working Group of the Mechanical Tests of Lubricants Panel of the Institute of Petroleum were used. These are : a tip relief of 0·0002 to 0·0004 inch on the fifteen-tooth driving gear and of 0·0004 to 0·0006 inch on the sixteen-tooth driven gear, with the relief in both cases starting from approximately the highest point of single-tooth contact; the remainder of the active part of the profile to be within $\pm 0·0001$ inch of true involute.

For all tests except the comparison of different materials, the gears

Table I

Basic Dimensions of Test Gears

	Fifteen-tooth driving gear	Sixteen-tooth driven gear
Face width (above pitch circle)	$\frac{3}{16}$ inch	
Diametral pitch (nominal)	5	
,, ,, (working)	4·7692	
Pressure angle of shaper cutter	20°	
Pressure angle, working, ψ	26° 19′ 16″	
Centre distance	$3\frac{1}{4}$ inch	
Base circle diameter	2·8191 inch	3·0070 inch
Pitch circle diameter	3·1452 inch	3·3548 inch
Root diameter	2·686 inch	2·896 inch
Tip diameter (as ground)	3·540 inch	3·750 inch

For a lever load W,

Torque in fifteen-tooth driving-gear shaft = $1·5W$ lb-ft.

Specific tooth load (tangential), F_t = $61·0W$ lb/in.

At first single-tooth contact

$$S_c * = \frac{F_t \sec \psi}{R_r^{0·8}} = 2·830\ F_t = 172·6W \text{ p.s.i. units}$$

$$P_{max} * = 2250\sqrt{S_c} = 32800\sqrt{W} \text{ p.s.i.}$$

* See Merritt,[1] p. 228.

were made from case-hardened nickel–molybdenum steel corresponding to BS En.34, or, in earlier days, its American counterpart, SAE 4615. The hardness was usually between 720 and 780 V.P.N. after grinding, but in some cases it was as high as 850 V.P.N. or more. In general, each particular investigation was made on a single batch of gears to eliminate the effect of variations of this kind.

In Table I are given the basic design dimensions of the test gears, as well as certain conversion factors, for comparison with the contact conditions of other types of test specimens.

Test Procedure

The method of test used has been uniform throughout, except where, as indicated by the type of test, certain factors, such as speed, temperature, etc., have become the variables instead of the constants.

The " standard " test conditions have involved running the machine at a shaft speed of 2000 r.p.m. of the fifteen-tooth driving gear (1875 r.p.m. of the sixteen-tooth driven gear), the lubricant being supplied to the point of entry into mesh, through a $\frac{1}{32}$-inch orifice at the rate of $\frac{1}{2}$ pint/minute. For most tests, other than those on the effect of temperature, the lubricant temperature was maintained at 60° C, though in some a temperature of 90° C was used.

The general method was as follows : beginning with an equivalent lever load of 10 lb (*i.e.*, 610 lb/inch face width on the test gears), and rising by increments of 5 lb, the gears were run for 5 minutes at each load until scuffing occurred within the 5-minute period. The time for inspection and re-loading was maintained constant at 5 minutes throughout each test.

The onset of scuffing was noticeable first by the decrease in the speed of the machine, and with the generation of white vapour in the test gearbox, and secondly by visual inspection after each 5-minute run. The general practice was to make several tests on any oil, or under any particular set of conditions, and to take the mean value of the scuffing load so obtained as the significant value.

In general, two tests could be made on each pair of gears, after which the scuffed gears could be reclaimed for a further two tests by having the profiles re-ground. Such re-ground gears were considered to form a new batch of specimens, and this reclaiming process was only done once on any particular pair of gears on account of the risk of unduly reducing the case thickness.

Figs 3 and 4 show photographs of the faces of an unused gear and a scuffed case-hardened gear, each taken normal to the tooth surface. It will be noted that the scuffing does not extend across the pitch region where there is no relative sliding. Fig 4 is typical of a gear scuffed in an undoped mineral oil.

Results of Gear-scuffing Tests

Effect of Lubricants not containing Additives

(i) *Viscosity.* Tests on plain mineral oils at five general viscosity levels,[2] obtained by blending light and heavy oils in different proportions, and by taking distillates of different boiling points from the same crude, showed

that, at a given temperature, the higher the oil viscosity the greater the protection against scuffing.

The range of scuffing loads for these oils, whose kinematic viscosities ranged from approximately 15 to 160 cs at the test temperature of 60° C was such that the heaviest carried a load about 3·2 times that of the lightest.

TABLE II

Compositions and Physical Properties of Oils Used for Tests on the Effect of Viscosity

Series	Oil no.	Approx SAE no.	Composition (percentages by weight)	Kinematic viscosity, cs		Viscosity index	Viscosity, Redwood I at 140° F, sec
				210° F	100° F		
I	1	10	Distillates from the same Venezuelan crude	5·9	37·2	111	71
	2	20/30		9·35	78·5	104	121
	3	40		14·5	162·0	96	219
	4	50		20·5	275·0	95	342
II	5	20/30	No. 1 + 34% No. 15	10·3	86·8	108	130
	6	40/50	No. 1 + 62% No. 15	15·5	164·6	103	219
	7	50/60	No. 1 + 77% No. 15	21·2	267·3	102	332
III	8	20	No. 1 + 32% No. 16	9·66	77·4	111	128
	9	40/50	No. 1 + 59% No. 16	15·7	165·7	104	220
	10	50/60	No. 1 + 73½% No. 16	20·6	260·8	101	318
IV	11	10	150 Penna neutral	5·4	34·3	100	65
	12	20/30	No. 11 + 37½% No. 15	10·1	88·3	103	130
	13	40/50	No. 11 + 64% No. 15	15·9	179·4	99	229
	14	50/60	No. 11 + 78% No. 15	21·0	263·3	100	321
	15	140	Penna bright stock	33·8	556·2	99	625
	16	140	Penna dark cyl stock	35·3	735·0	85	649

In fact, it was found that the scuffing load varied closely with the square root of the kinematic viscosity for this particular group of lubricants and for the design of spur gears used.

The compositions and physical properties of the oils used in this investigation are given in Table II, and the results are shown graphically in Fig 5.

Tests have also been made on a few oils of viscosity rather lower than the lower limit quoted above—oils of the DTD 44 D type commonly used in aircraft hydraulic systems, and in gas-turbine engines. These have generally failed at slightly lower loads than the lightest oil quoted above, and their successful use for gear lubrication would appear to depend very largely upon the use of suitable additives.

These comments on the effect of oil viscosity on gear scuffing refer to oils having different viscosities at a given temperature, the increase in scuffing protection afforded by the heavier oils being possibly due to the presence of larger hydrocarbon molecules.

(ii) *Temperature.* Variations in viscosity caused by variations in the temperature of a given oil have been shown to have considerably less effect on scuffing than the use of different oils having equivalent differences in viscosity at some given temperature.

An investigation of the effect of supply temperature [3] on scuffing of

mineral oils of various types (paraffinic, naphthenic, and highly refined naphthenic white oils), indicated that different oils reacted in different ways to variation in oil-supply temperature.

The paraffinic oils and white oils gave very little change in scuffing load over the range 40° to 110° C, while of two naphthenic oils the heavier showed very little change, but the lighter showed a marked reduction in scuffing load with increase in temperature. The naphthenic oils gave somewhat greater scatter of results than the others.

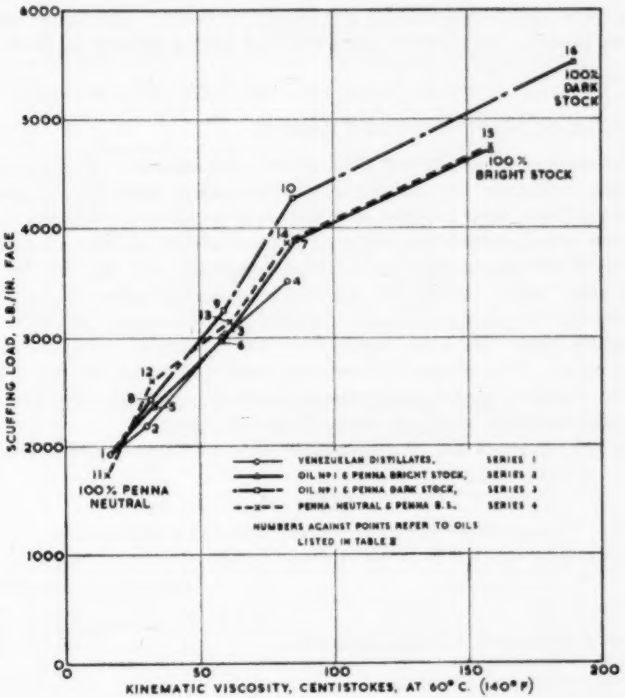
SCUFFING LOAD, LB./IN. FACE
KINEMATIC VISCOSITY, CENTISTOKES, AT 60° C. (140° F)
100% DARK STOCK
100% BRIGHT STOCK
100% PENNA NEUTRAL
VENEZUELAN DISTILLATES, SERIES 1
OIL Nº1 & PENNA BRIGHT STOCK, SERIES 2
OIL Nº1 & PENNA DARK STOCK, SERIES 3
PENNA NEUTRAL & PENNA B.S., SERIES 4
NUMBERS AGAINST POINTS REFER TO OILS LISTED IN TABLE II

FIG 5

EFFECT OF VISCOSITY ON SCUFFING LOAD (FOR DETAILS OF OILS SEE TABLE II)

(iii) *Chemical Nature of Oil.* In the tests described under (i) on the effect of oil viscosity, it was found that those oils at the heavier end of the range which consisted mainly or entirely of Penna dark stock had slightly better anti-scuffing properties than those which consisted mainly or entirely of Penna bright stock. The heaviest of four Venezuelan distillate oils (SAE 50) did not give quite such good protection against scuffing as blended oils of equivalent viscosity containing about 77 per cent of Penna bright stock suitably diluted with low-viscosity distillates.

The investigation mentioned under heading (ii), in which oils of various types were tested at different temperatures, indicated that, of three oils

having approximately the same viscosity at 60° to 70° C (their viscosity indexes varied), and acid-refined naphthenic oil gave greater protection against scuffing than a blended paraffinic oil, while the latter was appreciably better than a highly acid-refined solvent extracted naphthenic white oil.

The only non-mineral oil for which information is available is castor oil. Tests on a proprietary racing oil of which castor oil was the main constituent [4] gave a rating of about 1·6 compared with an SAE 50 solvent-refined mineral oil, but a further series of tests made at a later date gave to treated castor oil (to specification DTD 72) a rating of 3·33 compared with a mineral oil to specification DTD 109 (about SAE 50). The latter tests were, however, made at a higher temperature and with a greater oil flow than the former.

Effect of Lubricants containing EP additives

(i) *Advantages to be Gained from Various Additives.* A number of oils containing additives of various kinds, including proprietary ones, have been tested from time to time, and the main purpose of selecting data from these tests is to illustrate the order of improvement which can be gained.

Additives for improving the scuffing properties of oils can be broadly divided into " mild " additives, which give a small degree of improvement, and powerful " extreme-pressure " additives which may prevent scuffing up to many times the load required to scuff gears in the presence of the base oil only. The additive is usually restricted to the milder type when the oil is required to lubricate mechanisms in addition to gears, *e.g.*, an aero-engine complete with air-screw reduction gears.

Table III [4] gives a list of the ratings of a number of typical additives.

TABLE III

Comparative Ratings of Various Additives in Mineral Oil

Additive	Base oil	Rating
None	SAE 50 mineral oil	1·0
2·5% conc suspension of colloidal graphite in oil	" "	1·1
1% methyl dichlorstearate	" "	1·3
0·25% naphthalene tetrachloride	" "	1·5
0·5% naphthalene tetrachloride	" "	1·6
0·5% proprietary additive A *	" "	1·6
1% proprietary additive A *	" "	1·7
1% tricresyl phosphate	" "	2·0
1% thio-ether compound	" "	2·1
1% thio-ether phosphite	" "	2·3
5% proprietary additive B †.	" "	>2·9
5% sulpho-chloro compound	" "	>2·9
10% sulpho-chloro compound	" "	>2·9

* Proprietary additive A contained 80% tricresyl phosphate and 20% tributyl phosphite.

† Proprietary additive B was a chlorinated wax.

Available data concerning other proprietary additives of the very mild type have been omitted as being of little interest.

It will be seen that additives can increase the load at which scuffing occurs with ordinary engine oils by, at least, about three times.

(ii) *Response of an Oil to Varying Concentration of an Additive.* As an illustration, results are given in Table IV, of the effect of adding various concentrations, ranging from $\frac{1}{4}$ to 4 per cent, of tricresyl phosphate to a mineral engine oil to specification DTD 109. It will be seen that concentrations above about 2 per cent yield little further improvement. This result applies only to this particular combination of additive and oil, but shows the order of effect which may be expected.

TABLE IV

Response of an Oil to Varying Concentration of an Additive

Base Oil to Spec DTD 109. Additive : Tricresyl phosphate

Percentage of additive in the oil	Rating : $\frac{\text{doped oil}}{\text{base oil}}$
0	1
$\frac{1}{4}$	1·37
$\frac{1}{2}$	1·54
1	1·63
2	2·03
4	2·29

TABLE V

Response of Various Oils to the Same Additive

Additive : 1% tricresyl phosphate

Base oil	Rating (based on DTD 109 Source 1 as unity)	
	Undoped	Doped
DTD 109 :		
Source 1	1·00	1·87
Source 2	1·09	1·87
Source 3	1·26	1·74
Source 4	1·35	2·00
DTD 472	1·83	2·35

(iii) *Response of Various Oils to the Same Additive.* Table V shows the results obtained on each of five medium-viscosity engine oils, four of them to specification DTD 109 and the fifth to specification DTD 472, obtained from as widely different sources as possible, and on a sample of each of these oils containing 1 per cent tricresyl phosphate. This table, which gives the relative ratings, based on DTD 109 Source 1 as unity, shows : (*a*) how the scuffing properties of oils nominally to the same specification can vary, and (*b*) how the poorer oils respond to the additive to a greater degree than the good oils. It should again be emphasized that the effects illustrated by these results may be of different magnitudes with other oils and additives.

Effect of Gear-tooth Material and Nature of Surfaces

(i) *Gear Material and its Heat Treatment.* In an investigation of the effect of gear material and heat treatment on failure by scuffing,[5] the scuffing loads of gears made from a variety of gear steels, including case-hardened, through-hardened, and nitrided types, were determined, using both a medium-viscosity undoped mineral oil and a low-viscosity oil containing a mild EP additive as lubricants.

The following conclusions were drawn :—

(*a*) There was no systematic variation of scuffing load with surface hardness over the range 360 to 840 V.P.N., though there was a trend towards an increase in scuffing load with increasing hardness with the low-viscosity doped oil.

(*b*) Within the range of materials tested, high nickel content appeared to be a disadvantage, whereas both chromium and molybdenum were beneficial.

(*c*) The effect of the microstructure of the case-hardening steels tested was conflicting. A coarse acicular martensitic structure gave higher scuffing loads in two instances with the medium-viscosity undoped oil than a fine (re-quenched) structure in the same steel, but the finer structure, which had a higher hardness, gave greater protection against scuffing in the low-viscosity doped oil.

(*d*) Nitrided steel was very resistant to scuffing, and gave the highest scuffing loads of all materials examined.

(*e*) The presence of 0·2 per cent of lead in a steel (for improved machinability) made no significant difference to the scuffing properties of the material.

(*f*) Combinations of different materials failed at loads intermediate between the failure loads of the individual materials.

(*g*) Tests on gears made from forged blanks and from blanks cut from bar stock, all manufactured and heat treated under identical conditions, indicated that the macro-structure of the gear blank had no influence on the scuffing load.

(iv) *Chemical Surface Treatments.* It has been shown [6] that the resistance to scuffing of a pair of gears in a given oil may be increased by the application of certain chemical treatments to the surfaces of the gears.

The treatments were applied to the teeth of ground gears, and the gears were tested for scuffing with a medium-viscosity mineral oil. Certain treatments, such as caustic-soda–sulphur treatment and the anodic deposition of colloidal graphite, were found to be ineffective. The ratings of the various treatments which showed some promise are given in Table VI compared with untreated gears. Ratings are also given for gears which were first " run-in " for 100 hr at a load just below the scuffing load of untreated gears, and which were then subjected to the usual scuffing test. This was done to determine whether the surface treatments were likely to wear off.

The general conclusion from these tests was that the ferrous and manganese phosphate treatment was the most effective, approximately doubling the load-carrying capacity of the gears, and that it would withstand prolonged running without deterioration.

(v) *Electro-deposited Metallic Platings.* Scuffing tests on gears treated with a variety of electro-deposited metallic platings [7] have shown that certain of these are beneficial. Results are given in Table VII, and show the

TABLE VI

Effect of Chemical Surface Treatments

Surface treatment	Rating : $\frac{\text{treated}}{\text{untreated}}$	Rating after 100 hr running-in : $\frac{\text{treated}}{\text{untreated}}$
Untreated gears	1	1
Ferrous and manganese phosphate; deep high-temp treatment	2·1	2·0
Ferrous and manganese phosphate; accelerated treatment	2·1	1·9
Ferrous and zinc phosphate; shallow medium temp treatment	1·3–1·5	1·2–1·5
Above treatment prolonged	1·7	1·2
Ferrous and zinc phosphate; low-temp shallow treatment	1·6	1·1

rating of a pair of plated gears relative to a pair of ground gears unplated. Owing to scarcity of specimens, very few tests were made on gears of each type, but the results show sufficiently large variations between many of the platings for them to be regarded as significant.

TABLE VII

Effect of Electro-deposited Metallic Platings

Type of plating	Rating : $\frac{\text{plated}}{\text{unplated}}$	Rating after 100 hr running-in : $\frac{\text{plated}}{\text{unplated}}$
Silver, soft, on top of copper	>2·5	2·5
Silver, hard, direct on steel	2·0	2·0
Silver, soft, direct on steel	2·0	2·1
Cadmium	1·7	1·4
Chromium, hard	0·4	—
Chromium, soft	0·4	—
Cobalt	1·5	1·3
Copper, hard	1·8	1·4
Copper, soft	2·0	1·8
Lead	1·2	1·0
Nickel	1·5	1·6
Tin	>2·1	2·0
Zinc	1·5	1·0
Brass	1·9	1·5
Bronze	2·2	2·3
Speculum	2·3	2·3

As with the chemical surface treatments results are also shown for gear pairs which had had a prior run of 100 hr at a load just below the failure load of unplated gears. It will be seen that the majority of the plated specimens retained their efficacy.

In general, it was found that silver, tin, bronze, and speculum platings

were the most effective in inhibiting scuffing. The next best, of those tested, was a soft deposit of copper. Hard copper, brass, and cadmium were inferior, especially after prolonged running. Nickel and cobalt platings gave slight protection, but zinc gave none at all. Chromium, whether a hard or soft deposit, disintegrated at loads only half as great as the reference load for unplated gears. The influence on scuffing did not appear to be affected by plating thickness within the approximate range 0·00005 to 0·0003 inch.

Effect of Running Conditions

(i) *Speed.* In tests made on En.34 case-hardened steel gears within the speed range 1000 to 4000 r.p.m., of the smaller gear corresponding to a

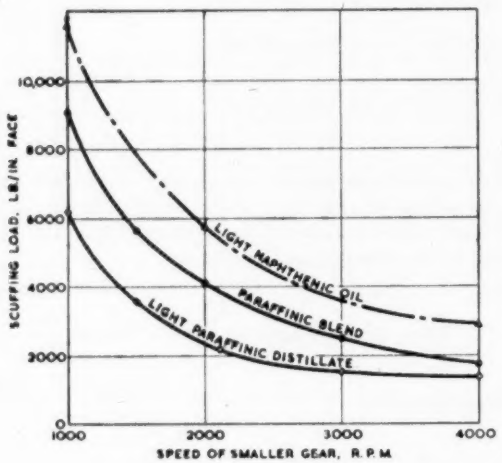

FIG 6

EFFECT OF SPEED ON SCUFFING LOAD FOR THREE OILS (PITCH LINE VELOCITY IN FT/MIN = 0·824 × SPEED OF SMALLER GEAR)

pitch-line velocity range of about 14 to 55 ft/sec and a gear-tooth sliding velocity range of 6·5 to 26 ft/sec at the tips in approach, there was a substantial change of scuffing load.

For each of three mineral oils, two of which differed widely in viscosity, and two in type, the scuffing load decreased with increase in speed, being proportional to $1/v^n$, where v is the sliding velocity and n is slightly greater than unity for the oils and particular gears used. The above 1 : 4 range in speed thus gave a range in scuffing load of approximately 5 : 1, as may be seen from Figs 6 and 7.

A different relationship was, however, obtained when the lowest viscosity oil of the above three was doped with tricresyl phosphate, the curve of scuffing load against speed tending to approach that of the base oil towards the ends of the range. No other information has been obtained on the behaviour of additives in relation to speed.

(ii) *Method of Supplying Lubricant.* Incidental observations made in the

early development of the technique of scuffing testing and at various later times, show that scuffing failure is unaffected by the change in position

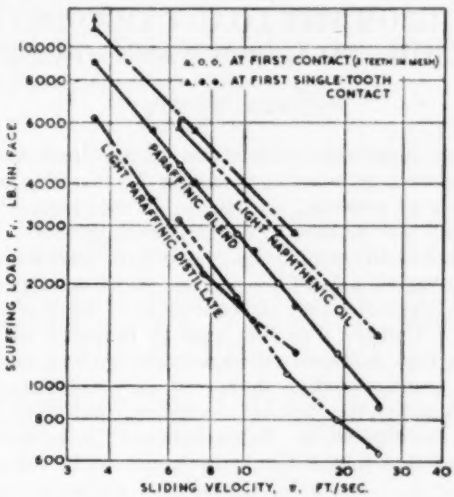

FIG 7

RELATIONSHIP BETWEEN SCUFFING LOAD AND SLIDING VELOCITY (BASED ON CURVES IN FIG 6)

of the lubricating-oil jet from the ingoing to the outgoing side of the gear mesh.

ACKNOWLEDGMENTS

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THE EFFECT OF VARIATION OF VISCOSITY WITH PRESSURE ON THE LOAD-CARRYING CAPACITY OF THE OIL FILM BETWEEN GEAR-TEETH

By Ewen McEwen *

There have been numerous calculations of the load-carrying capacity of the oil film between gear-teeth, assuming the viscosity of the lubricant to be independent of pressure, and there are two papers [3, 6] which have taken into account the increase in viscosity with pressure. Both of these, however, make use of the Sommerfeld boundary condition that no load is carried by the divergent part of the oil film, which has been shown [2] to be incorrect, since it does not satisfy the conditions of continuity and minimum potential energy. Further, a recent paper by Borsoff *et al* [1] objects to the lack of a limiting film thickness criterion in the hydrodynamic theory, and suggests that it be discarded in favour of an empirical approach. The present paper is based on the Reynolds boundary conditions, and endeavours to meet some of the objections. In particular, it is the aim of the present discussion to show that the oil-film capacity cannot be the cause of the use of an index lower than unity for the relative radius of curvature in the Hertzian load-carrying capacity of gear-teeth with film lubrication.[7]

Reynolds' differential equation for the pressure in an oil film is well known to be :—

$$\partial/\partial x \,.\, (h^3/\mu) \,.\, \partial p/\partial x - 6v\partial h/\partial x = 0 \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where h is the film thickness at x, p the pressure in the film at x, μ the oil viscosity, and v the relative velocity of the surfaces (taken positive in the opposite direction to x). In the absence of side leakage the partial differentials with respect to x can be written as full differentials and the equation integrated. The assumption of no side leakage can be justified for gear-teeth, as the axial length of the contact band is many times its effective width. Equation (1) then becomes

$$\{1/(6\mu v)\}dp/dx = (1 - C/h)/h^2 \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where C is a constant of integration.

Since the width of the contact band is narrow compared with the relative radius of curvature

$$h = h_0 + x^2/2r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where h_0 is the minimum film thickness at $x = 0$ on the common normal to the surfaces and r is the relative radius of curvature.

Let ϕ be a parameter defined by :

$$\cos^2 \phi = h_0/h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

then $$x = \tan \phi \,.\, \sqrt{2rh_0}; \; dx = \sec^2 \phi \,.\, \sqrt{2rh_0} \,.\, d\phi \quad . \quad . \quad (5)$$

Substituting in equation (2)

$$\{1/(6\mu v)\}(h_0^2/\sqrt{2rh_0})dp/d\phi = \cos^2 \phi(1 - \cos^2 \phi \,.\, C/h_0) \quad . \quad . \quad (6)$$

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One boundary condition is that $p = 0$ at $x = -\infty$; further, since the pressure is never negative in the oil film there is at some value of $x = +a$ zero pressure and also dp/dx zero. If α is the value of ϕ at $x = +a$, C/h_0 must be $\sec^2\alpha$. Equation (6) then becomes

$$\{1/(6\mu v)\}(h_0^2/\sqrt{2rh_0})dp/d\phi = \cos^2\phi(1 - \sec^2\alpha\cos^2\phi) \quad . \quad . \quad (7)$$

Assuming a pressure viscosity relation of the form

$$\mu = \mu_0(1 + p/k)^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

which satisfies most lubricants over a reasonable range by suitable choice of n and k [4, 5] we have

$$\{1/(6\mu_0 v)\}(h_0^2/\sqrt{2rh_0})\{1/(1 + p/k)^n\}dp = \cos^2\phi(1 - \sec^2\alpha\cos^2\phi)d\phi \quad . \quad (9)$$

which can be integrated to obtain

$$\{1/(6\mu_0 v)\}(h_0^2/\sqrt{2rh_0})\{k/(n-1)\}\{(1 + p_0/k)^{1-n} - (1 + p/k)^{1-n}\}$$
$$= \tfrac{1}{8}\sec^2\alpha\{(4\cos^2\alpha - 3)(\phi + \sin\phi\cos\phi) - 2\sin\phi\cos^3\phi\}. \quad (10)$$

since at $x = -\infty$, $\phi = -\frac{1}{2}\pi$, $\cos\phi = 0$, $p = 0$

$$\{1/(6\mu_0 v)\}(h_0^2/\sqrt{2rh_0})\{k/(n-1)\}\{1 - (1 + p/k)^{1-n}\}$$
$$= \tfrac{1}{8}\sec^2\alpha\{(4\cos^2\alpha - 3)(\tfrac{1}{2}\pi + \phi + \sin\phi\cos\phi) - 2\sin\phi\cos^3\phi\}. \quad (11)$$

Again since $p = 0$ at $x = +a$, $\phi = +\alpha$

$$\tfrac{1}{8}\sec^2\alpha\{(4\cos^2\alpha - 3)(\tfrac{1}{2}\pi + \alpha + \sin\alpha\cos\alpha) - 2\sin\alpha\cos^3\alpha\} = 0. \quad (12)$$

The solution to which is

$$\alpha = 0{\cdot}4436 \text{ radians} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Define $$\xi = \tfrac{3}{4}(n-1)\pi(4 - 3\sec^2\alpha)(\mu_0 v/k)(\sqrt{2rh_0}/h_0^2) \quad . \quad . \quad . \quad (14)$$

Then $$p/k = -1 + (1 - \{\xi/\pi(4\cos^2\alpha - 3)\}\{(4\cos^2\alpha - 3)(\tfrac{1}{2}\pi + \phi + \sin\phi\cos\phi) - 2\sin\phi\cos^3\phi\})^{1/(1-n)} \quad . \quad (15)$$

This has a maximum at $\phi = -\alpha$ which becomes infinite if

$$\{\xi/\pi(4\cos^2\alpha - 3)\}\{(4\cos^2\alpha - 3)(\tfrac{1}{2}\pi - \alpha - \sin\alpha\cos\alpha) + 2\sin\alpha\cos^3\alpha\} = 1 \quad . \quad . \quad (16)$$

But

$$\{\xi/\pi(4\cos^2\alpha - 3)\}\{(4\cos^2\alpha - 3)(\tfrac{1}{2}\pi + \alpha + \sin\alpha\cos\alpha) - 2\sin\alpha\cos^3\alpha\} = 0 \quad . \quad . \quad (17)$$
(from 12)

Adding, $\{\xi/\pi(4\cos^2\alpha - 3)\}\{\pi(4\cos^2\alpha - 3)\} = 1$, *i.e.*, $\xi = 1$ . . (18)

Now the total load carried by the film per unit axial length of gear-tooth is

$$P = \int_{-\infty}^{a} p dx = \int_{-\pi/2}^{\alpha} \sqrt{2rh_0}\sec^2\phi \,.\, p \,.\, d\phi \quad . \quad . \quad (19)$$

$$P/(k\sqrt{2rh_0}) = \int_{-\pi/2}^{\alpha} \sec^2\phi\{-1 + (1 - \xi\{\tfrac{1}{2} + (\phi + \sin\phi\cos\phi)/\pi - 2\sin\phi\cos^3\phi/(4\cos^2\alpha - 3)\pi\})^{(1/1-n)}\}d\phi = J \quad . \quad (20)$$

where J is the value of this integral.

Multiply both sides by

$$h_0^2 k/(\mu_0 v\sqrt{2rh_0}) = \tfrac{3}{4}(n-1)\pi(4 - 3\sec^2\alpha)/\xi \quad . \quad . \quad . \quad (21)$$

$$Ph_0/(2\mu_0 vr) = \tfrac{3}{4}(n-1)\pi(4 - 3\sec^2\alpha)J/\xi \quad . \quad . \quad . \quad . \quad (22)$$

$$\left.\begin{aligned} P &= (2\mu_0 vr/h_0)\tfrac{3}{4}(n-1)\pi(4 - 3\sec^2\alpha)J/\xi \\ &= (2\mu_0 vr/h_0)A \quad . \quad . \quad . \quad . \quad . \quad . \quad . \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (23)$$

where $A = \tfrac{3}{4}(n-1)\pi(4 - 3\sec^2\alpha)J/\xi$.

Now for no pressure viscosity effect,[7] $P = 2{\cdot}448\mu_0 vr/h_0$

$$i.e.,\ A = 1{\cdot}224 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

$$P = 1{\cdot}224(2\mu_0 vr/h_0)B \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where $B = \tfrac{3}{4}(n-1)\pi(4 - 3\sec^2\alpha)J/(1{\cdot}224\xi)$ and $B = 1{\cdot}0$ when the viscosity is independent of pressure.

Values of B obtained by numerical integration of equation (20) are given in Table I for $n = 6$, 8, 11 and ξ from 0 to 1.

TABLE I

Values of B *for Total Load* $P = 1{\cdot}224\ (2\mu_0 vr/h_0)B$

	$n = 6$	$n = 8$	$n = 11$
$\xi = 1{\cdot}0$	2·802	2·584	2·439
0·9	1·913	1·837	1·784
0·8	1·640	1·594	1·562
0·6	1·362	1·346	1·325
0·4	1·202	1·192	1·184
0·2	1·092	1·088	1·085
0	1·0	1·0	1·0

The values of B in Table I can be compared with the Hersey and Lowdenslager [6] values based on the Sommerfeld conditions for the critical case of $\xi = 1$, as shown in Table II. The line $1{\cdot}224B$ is because the Reynolds

TABLE II

Comparison of Critical Values of B *on Reynold and Sommerfeld Conditions*
$\xi = 1{\cdot}0$

	$n = 6$	$n = 8$	$n = 11$
Reynolds from Table I .	2·802	2·584	2·439
$1{\cdot}224\ B$ from Table I .	3·429	3·162	2·985
Sommerfeld (Hersey [6]) .	3·60	3·115	2·95

conditions gives a load of 1·224 as great as the Sommerfeld in the case viscosity *independent* of pressure.

Maximum Hydraulic Pressure

The singularity of the integrand of equation (20) clearly represents a limiting case, as in all real circumstances the hydraulic pressure must remain finite. Any value of ξ in excess of unity requires negative pressures and has no physical meaning. A limiting value of the load is obtainable with ξ unity for all n greater than 3, as in these cases the integral of equation (20) remains bounded.

From the limiting value of ξ we ascertain that hydrodynamic lubrication will break down if

$$\xi \geqslant 1, \textit{ i.e., } h_0 \leqslant \{2(\tfrac{3}{4}\pi\{4 - 3\sec^2\alpha\}^2 r(\{n-1\}\mu_0 v/k)^2\}^{1/3}$$

The minimum film thickness is then $h_0 = 1{\cdot}050(r\{(n-1)\mu_0 v/k\}^2)^{1/3}$ (26)

This gives a criterion for minimum film thickness.

The corresponding value of h_0 minimum, from Blok's discussion of Hersey [6], h_H is obtained by substituting $\sqrt{3}$ for $\pi(4 - 3\sec^2\alpha)$ in equation (26). This leads to a minimum film-thickness ratio for the two cases of

$$(h_0/h_H) = (\{\pi(4 - 3\sec^2\alpha)\}^2/3)^{1/3} = 0{\cdot}70 \quad . \quad . \quad . \quad (27)$$

In the limit, when the film thickness is at its minimum given by equation (26) we obtain by substitution in equation (25)

$$P = C_1 k^{2/3} \,.\, \mu_0^{1/3} \,.\, v^{1/3} \,.\, r^{2/3} \,. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

where $C_1 = 1{\cdot}224 \times 2 \times B_1/\{1{\cdot}05(n-1)\}^{2/3}$ (see Table III).

TABLE III

Values of C_1 *for Equation* (28)

	$n = 6$	$n = 8$	$n = 11$
C_1	2·271	1·673	1·240

The apparent drop in load-carrying capacity as n increases due to the lower values of C_1 is not really there, since a high exponent n to the pressure–viscosity relationship implies a high value of the viscosity constant k, which also appears in equation (28).

This leads to the conclusion that within the hydrodynamic range, the load-carrying capacity for a given value of $(\mu_0 v/h_0)$ increases rather more rapidly than the relative radius of curvature, since the value of B in equation (25) increases with r, until we reach the stage when the minimum hydrodynamic film thickness, from equation (26), is equal to that in the oil film. After this, any further increase in load-carrying capacity for a given value of $\mu_0 v$ necessitates a corresponding increase in both r and h_0 so that the net gain is only as the two-thirds power of the relative radius of curvature (equation (28)). At this stage the load-carrying capacity increases very slowly, only as the cube root, with $\mu_0 v$.

Where the minimum hydrodynamic film thickness given by equation (26) is less than that required from other considerations, such as surface finish, hydrodynamic lubrication will cease and boundary lubrication supervene at this earlier limit.

In practice, the difficulties of getting the lubricant to the working surface set a limit on the maximum value of $\mu_0 v$, so that for a given relative radius of curvature and sliding velocity the minimum film thickness and maximum load are both fixed. Any attempt further to increase $\mu_0 v$ may merely eliminate hydrodynamic lubrication, and consequently it is not surprising that Borsoff *et al* [1] found that the maximum scuffing horsepower tends to become independent of $\mu_0 v$ for given gears. The anomalous behaviour of silicones and glycerine found by them may, however, be at least in part due to the exceptionally small pressure-viscosity effect for these lubricants.[8]

The final result of this investigation is to suggest that gear-tooth-surface load-capacity formulæ should consist of two parts : (*a*) a region of fluid film lubrication in which surface failure is by pitting due to fatigue and (*b*) a region of boundary lubrication in which surface failure is by scuffing due to breakdown of the lubricant. The change-over limit from one region to the other is reached when the fluid film thickness attains the minimum permissible value determined either by the surface roughness or the hydrodynamic minimum here discussed; for surfaces of sufficiently high finish, therefore, the boundary between the regions is such that the maximum of the hydrodynamic zone increases as the two-thirds power of relative radius of curvature.

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[2] Cameron, A., and Wood, W. L. (Mrs). *Proc. I.Mech.E.*, 1949, **161**, 59.
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[5] Hersey, M. D. *Trans. A.S.M.E.*, 1951, **73**, 673.
[6] Hersey, M. D., and Lowdenslager, D. B. *Trans. A.S.M.E.*, 1950, **72**, 1035.
[7] McEwen, E. *Engineer*, 1948, **186**, 234, 667.
[8] Bridgman, P. *Proc. Amer. Acad. Arts Sci.*, 1949, **77**, 115.

NOTE ON PERFORMANCE OF GRAPHITED OIL

By E. A. Smith * (Member)

Graphite is a lubricant because it slips or cleaves with facility when rubbed. The weakness in shear is such as to constitute a high measure of lubrication. When rubbed, this solid lubricant gives rise to fresh, uncontaminated faces with high adsorptive energy, which enables them to hold oil and to adsorb to clean metallic faces. Mechanical strength in a direction normal to the shear planes is comparatively high, and this, in association with a high facility for slip in shear, enables graphite to offset metal pick-up between two rubbing faces.

When colloidal graphite in oil is used to lubricate the rubbing faces of mechanical equipment, some of the particles of graphite are spread out on the rubbing faces like microscopic cards. Under favourable conditions, it is possible to form, thereby, a thin, adsorbed film of graphite on the metal faces, which has been called, for convenience, a graphoid surface.

It is not the aim of these notes to discuss the low frictional character of the graphoid surface, but the volume of data now available from various sources, suggests that such a graphited surface offers a coefficient of friction lying between 0·10 and 0·13, for rubbing speeds up to 4000 ft/min and for loads up to the beginning of the extreme-pressure range. This coefficient should represent the maximum value to which metal rubbing faces of smooth finish should go under boundary conditions, when the oil-film is broken temporarily. When such a fluid film rupture does occur, the affinity for

* Acheson Colloids Ltd.

oil exhibited by a graphited surface encourages wetting of the exposed areas in the minimum time.

It is instructive to follow through the effect of graphited lubricants from laboratory tests on machines designed to bring out their lubricating and load-carrying qualities to actual gears, and below is described such a sequence of results.

A test machine which simulates the rolling and sliding contact in gear-teeth is that employing two hardened steel solid cylinders one running at 50 r.p.m. and the other at 300 r.p.m. The two cylinders are not parallel but run skew-wise against each other, so producing a sinusoidal line contact due to slight movement of one roller parallel to its axis of rotation. By suitable load arms on the cylinders or rollers, friction measurements can be made at various loads up to an estimated value of about 125,000 p.s.i. In such a testing machine, the coefficient of friction for graphited oil showed a progressively lower friction, compared with plain oil, up to 125,000 p.s.i. Comparative tests on an SAE mineral oil with and without colloidal graphite in a four-ball testing machine also reveal an increased load capacity, of 30 per cent, with colloidal graphite present.

Results on various other testing machines reinforce the general conclusions to be drawn from the above results. Turning now to the Timken machine, which takes into account wear and brings conditions nearer to gear practice, the results in Table I can be quoted.

TABLE I

	Load on lever, lb	Loss in weight of block and cups, g	Max load, lb
Cylinder oil	10	0·000	14
Cylinder oil plus 2% sulphur	10	0·0055	40
Cylinder oil plus 10% mica	10	0·2695	14
Cylinder oil plus 10% colloidal graphite	33	0·0330	85
Cylinder oil plus 10% dichloro-ethyl ether	33	0·0020	95

Here the fact emerges that graphited oil has high load-carrying qualities. It should be added that the results were selected from an assortment of data indicating that the oil carrier or the additive can play a big part in results.

Having established a *prima facie* case for graphited oil with testing machines, one can now turn to an actual gear test, in this instance, a worm gear. Results are shown in Table II, the " Average Increase " representing the improvement over a mineral or compounded oil, following the addition of colloidal graphite. Under the respective gear speeds are shown average increases of maximum load permissible with a mineral oil after the addition of colloidal graphite, expressed in percentage.

Here again, it should be interposed that the oil used influences results with colloidal graphite, minimum, and maximum gain of load-carrying capacity being shown. It might be added that the results quoted in Table II are in line with results obtained in tests reported by DSIR some years ago.

To sum up, it can be stated that colloidal graphite in a mineral lubricating oil can improve the load-carrying capacity of that oil, facilitating gear lubrication, where extreme pressures do not predominate. No claim is made for extreme-pressure qualities, but it is contended that a great deal of gear lubrication calls frequently for a moderate increase in lubricant

TABLE II

Lubricant	R.p.m. of worm				Average increase of load, %
	800	1000	1400	1800	
A	1·8	2·4	5·2	10·1	4·9
B	32·8	40·0	53·4	53·2	44·8
C	26·5	33·9	42·5	39·9	35·7

performance. There are certain specialized instances where graphited oil can perform a function not always possible with other lubricants, as for example, the reduction of certain pitch-line noises or facilitating running-in with minimum tooth wear. A higher critical temperature of gear-bath can also be sustained with colloidal graphite due to its ability to reinforce the weakened oil-film and to offset pick-up.

No reference has been made in the above notes to graphited greases. An industrial case for the use of such greases has been established, such lubricants having been standard practice for many years. It is imperative, however, to ensure that pure graphite is used and that this is dispersed homogeneously in the correct grease. Confusion has been caused, in the past, in evaluating graphited greases, due to the use of graphite of unknown specification.

THE WEAR AND PITTING OF BRONZE DISKS OPERATED UNDER SIMULATED WORM-GEAR CONDITIONS

By L. S. EVANS * and R. TOURRET * (Associate Fellow)

SUMMARY

A theory to account for the mechanism of wear under mixed hydrodynamic/boundary lubrication conditions is advanced, while the existing theories regarding the reasons for pitting are correlated and an attempt made to demonstrate that pits are liable to be initiated in practice by both internal and surface cracks.

It is suggested that a lubricant of highest viscosity consistent with other requirements without any chemical-type EP agent is desirable for bronze worm gears to give low wear and long life before pitting occurs.

INTRODUCTION

IT is intended to present here an appreciation of the present state of knowledge of wear and pitting of bronze surfaces operating under a loaded

* Thornton Research Centre, "Shell" Refining & Marketing Co. Ltd.

sliding-and-rolling contact. Such conditions exist in service in worm gears in automotive transmissions, etc.

In general it seems that metal of either a gear or a disk is removed in one of two ways :

(*a*) " Wear," * in which minute particles are continually removed, the surface remaining reasonably smooth.

(*b*) " Pitting," * in which comparatively large granules are removed intermittently, leaving distinct pits, to the detriment of the surface. (See Fig 1, which shows a typical pit shortly after its formation.)

During wear the amount of metal removed is very small and the surfaces remain usable, but during pitting the amount of metal removed is much greater and the surface rapidly deteriorates. The major practical need therefore appears to be to prevent or at least delay the onset of pitting. In some practical applications, however, pitting might not begin during the normal operating time between overhauls, and in these instances the rate of wear will be important.

A test rig has been built in order to permit the study of these phenomena, and this employs disks operated in loaded peripheral contact. By varying the speed and direction of rotation of the disks, as well as their diameter if necessary, combinations of sliding and rolling contact conditions experienced in service can be simulated within very wide limits. The advantages of using simulation-type rigs rather than rigs using actual gears [1, 2, 3] have long been realized, and many machines using two disks in contact to simulate gear conditions have been devised. The SAE Machine [4] was perhaps the first test rig employing disks as test pieces, but Way [5] and Merritt [6] soon had their designs in operation. These rigs were followed by those of Nishihara and Kobayashi,[7] Meldahl,[8, 9] and the Institution of Automobile Engineers Research Department.[3] The design of the Thornton disk wear machine is based upon that of the smaller 3-inch-centres disk machine of Merritt,[6] as developed at David Brown & Sons (Huddersfield) Ltd.[10] Some preliminary results from this rig, covering certain aspects of wear and pitting, will be given, together with a review of the ideas contained in the literature.

Thornton Disk Wear Machine

Description

Two disks are operated in loaded peripheral contact and driven either by gears or chain and sprockets to provide a range of sliding and rolling speeds. One disk is mounted on a shaft revolving in self-aligning bearings which are mounted in freely swinging housings so that the disks maintain an evenly loaded contact. The machine is driven by a constant-speed motor † and is fitted with automatic instrumentation for unattended

* These terms will be used only in the sense hereby defined.

† In contrast, a 3-inch-centre David Brown type disk machine, also operated at Thornton, is equipped for friction tests with a variable-speed swinging-field motor from which it is possible to determine the horsepower used, but has no automatic equipment. In view of these differences and the different functions of the rigs it has been deemed advisable at Thornton to refer to the new rig as the " disk wear machine." Both rigs will accommodate the same range of disks with the same range of gear or chain drives, such details being interchangeable between the two rigs.

operation. Oil is circulated at controlled flow and temperature by a pump through filters so that only clean oil is supplied to the contacting surfaces.

In the Thornton disk wear machine (Fig 2) it is customary to operate a 2-inch-diameter steel * and a 4-inch-diameter phosphor-bronze * disk together. Each disk is 1 inch wide. The bronze disk has three equally spaced contact lands each $\frac{3}{16}$ inch wide, while the steel disk has a plain cylindrical surface. A chain-and-sprocket coupling (Fig 3) is used in order to produce a motion simulating the conditions existing in bronze worm gears, namely, high sliding speed and the surfaces moving in opposite directions at the point of contact. The disks rotate at 750 r.p.m., and the sliding speed is 1180 ft/min, while the slide/roll ratios are 1·5 and 3 respectively, for the bronze and steel disks.

Test Technique

The technique employed for measuring wear is to make a very narrow file mark in the land of the bronze disk. The probe of a "Talysurf" surface meter is then run in an axial direction over this scratch, giving a trace from which the depth of the scratch can be determined, the bottom of the scratch providing a reference surface from which to measure the wear. This technique was considered likely to be more sensitive than using either an ordinary micrometer over the diameter of the disk, or a step micrometer straddling the lands and employing the two outer lands, of reduced diameter, as reference surfaces. The ability to produce to order scratches of perhaps between 0·0015 and 0·0020 inch depth is soon acquired. Readings are taken at convenient intervals, depending on the wear-rate, throughout the test. The results when totalled over a complete test agree reasonably well with the decrease in diameter as measured by a micrometer. With triple-land discs the mean depth of three scratches, one on each land, disposed at 120°, is used.

Wear and time-to-pitting tests are performed together, the test usually being concluded when the first pits occur. The disk is subjected to five hours running-in at the test load prior to the wear measurements. This running-in is regarded as the final stage in the preparation of the surface of the bronze disk for wear tests, but is included in the time-to-pitting results.

Disk Design

Originally the bronze disk was made with the middle land proud of the two outer lands to permit wear measurements with a step micrometer, only the middle land being in contact with the steel disk. The change to the scratch-and-"Talysurf" method of measuring wear removed the need for the outside reference lands and allowed the use of three lands simultaneously. This change increased the self-aligning couple sixteen-fold (Fig 4) and removed doubts as to whether the original self-aligning couple was sufficient (Fig 18, for instance, shows evidence of edge contact having occurred).

* See Appendixes I and II.

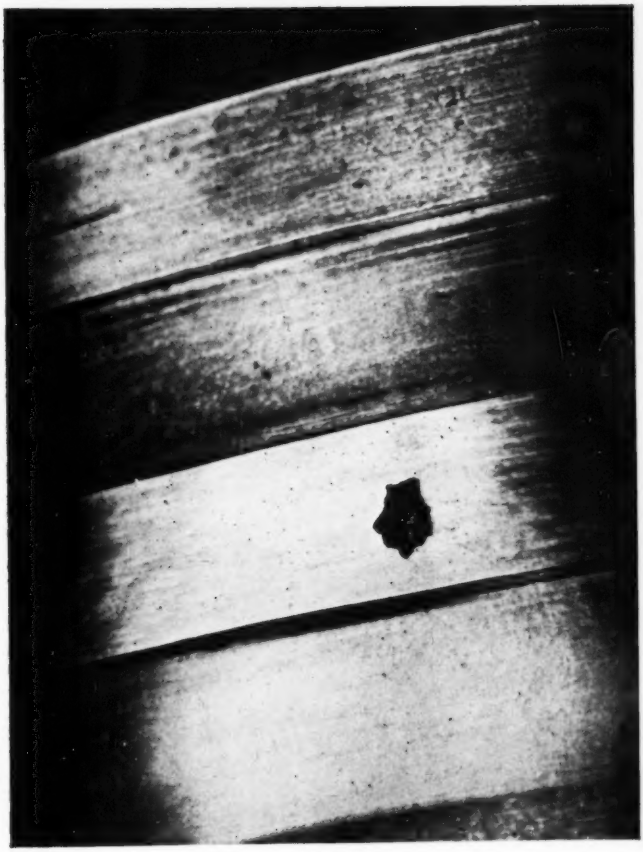


FIG 1

TYPICAL PIT

Photograph taken shortly after its formation

[*To face p.* 634.

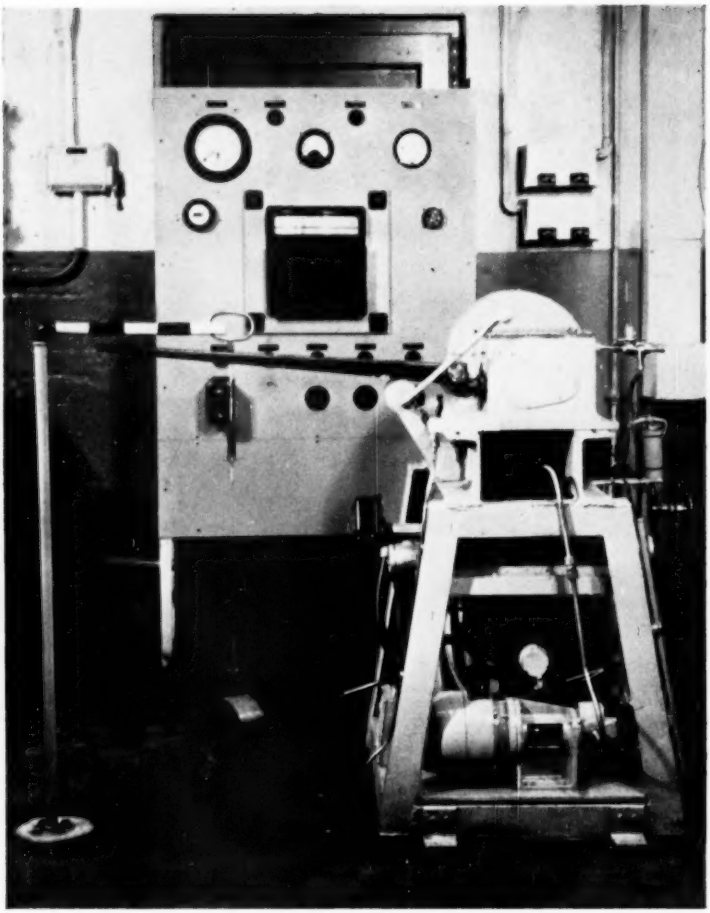


FIG 2

THORNTON DISK WEAR MACHINE

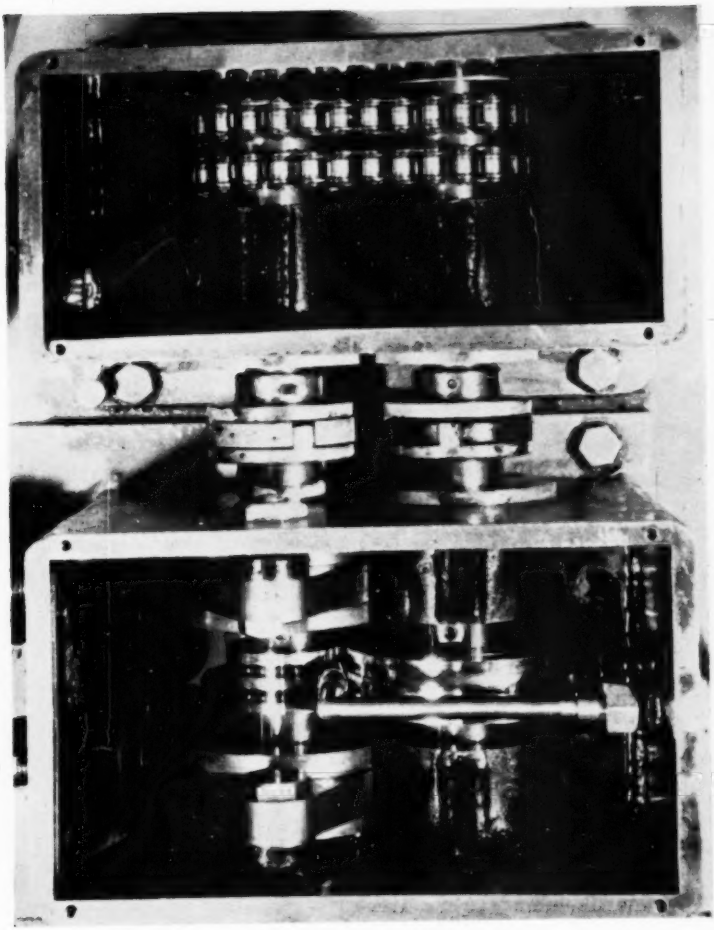


FIG 3

DISKS AND COUPLING ASSEMBLY OF THE THORNTON DISK WEAR MACHINE

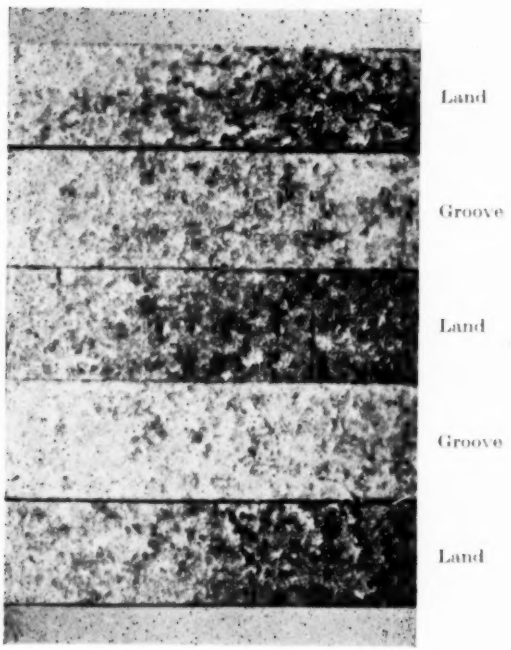


Fig. 9

GRAIN SIZE OF TYPICAL BRONZE DISK (×3)

FIG 12
HEAVILY PITTED DISK (TEST NO 13)

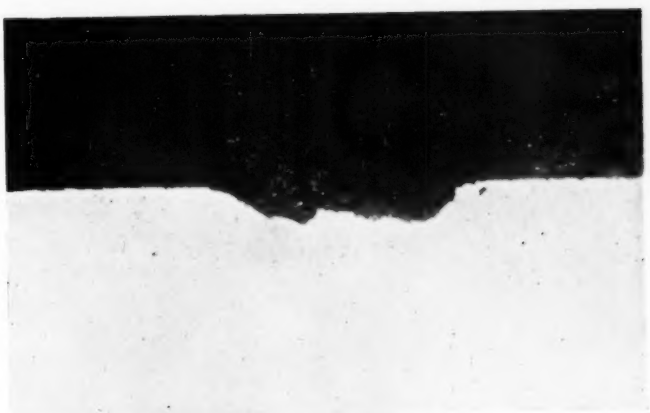


FIG 13
CROSS-SECTION OF SHALLOW PIT (× 40, MAX DEPTH 0·006 INCH)

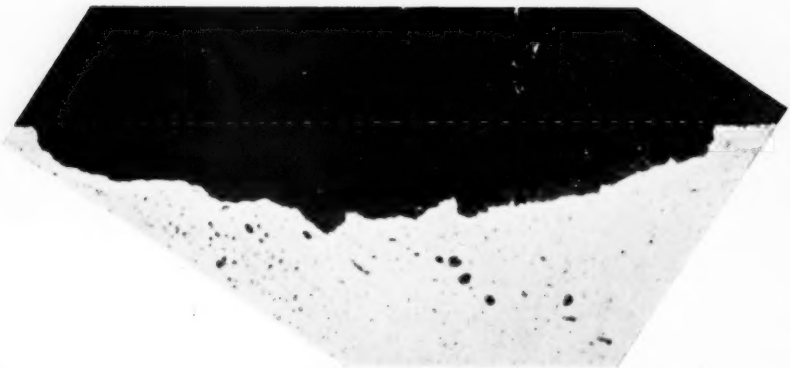


FIG 14

CROSS-SECTION OF DEEP PIT (×40, MAX DEPTH 0·015 INCH)

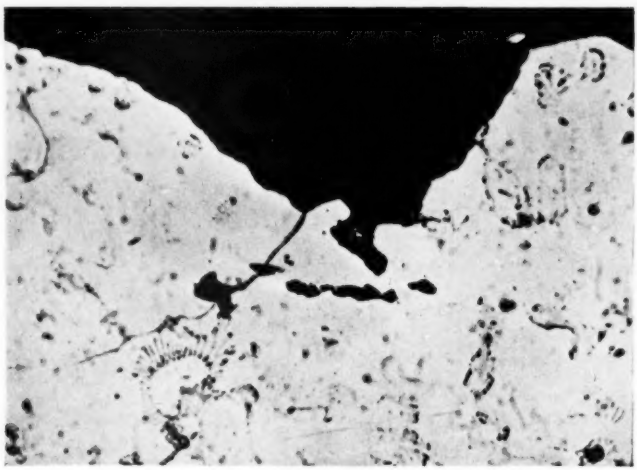


FIG 15

PROPAGATION OF CRACKS IN BOTTOM OF DEEP PIT (×300)

FIG 16

SUB-SURFACE CRACKS NEAR DEEP PIT (×300)

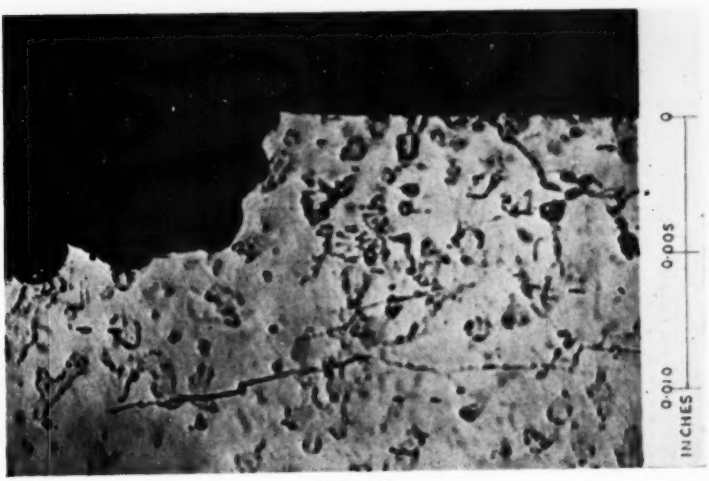


FIG 17

EDGE OF DEEP PIT (×150)

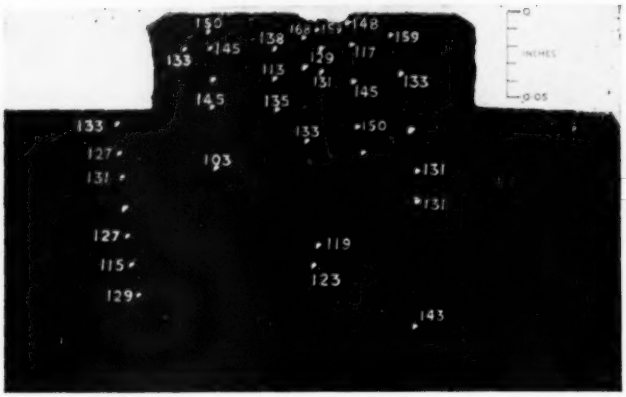


FIG 18

HARDNESS SURVEY OF SHALLOW PIT (NUMBERS RELATE TO V.P.N. DETERMINATIONS)

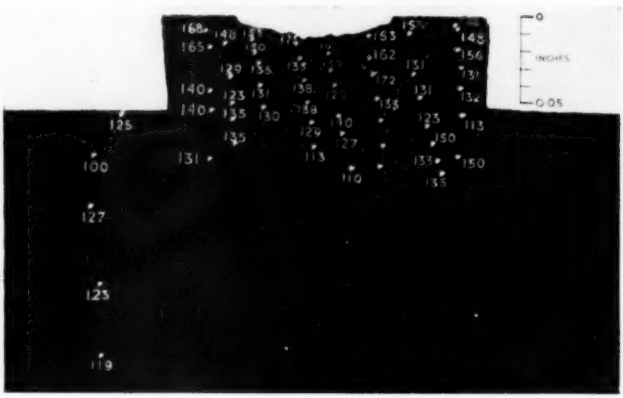


FIG 19

HARDNESS SURVEY OF DEEP PIT (NUMBERS RELATE TO V.P.N. DETERMINATIONS)

Mathematical Background

Let Fig 5 represent the two disks in loaded contact (with their axes parallel).

Let W = load, lb;

E = Young's modulus of elasticity, p.s.i.;

and l = length of disks in contact, inches;

so that p = load per unit length of line contact of cylinders, W/l.

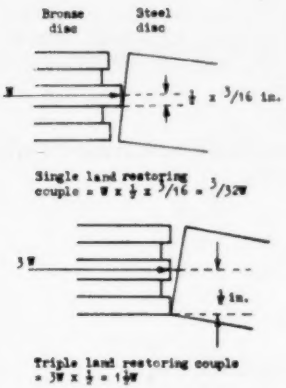
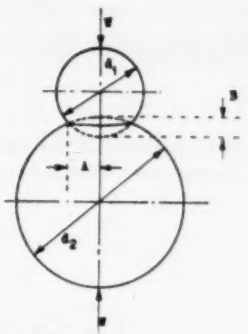

Bronze disc
Steel disc
W
½ x 3/16 in.
Single land restoring couple = W x ½ x 3/16 = 3/32W
3W
¼ in.
Triple land restoring couple = 3W x ½ = 1½W

FIG 4

SELF-ALIGNING COUPLES WITH SINGLE- AND TRIPLE-LAND CONTACTS



Let W = load, lb.
E = Young's modulus of elasticity, p.s.i.
l = length of cylinders in contact, inches.

FIG 5

DISKS IN LOADED CONTACT CONDITIONS

Following Hertz,[11] the maximum compressive stress in such a system (which is also the maximum pressure), p.s.i.

$$= 0{\cdot}591\sqrt{pE\left(\frac{d_1+d_2}{d_1d_2}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Further,

$$B \text{ (inches)} = \frac{1{\cdot}33p}{E}\left[7{\cdot}4 - \log_{10}\left\{\frac{p(d_1+d_2)}{2d_1d_2}\right\}\right] \quad . \quad . \quad (2)$$

and

$$A \text{ (inches)} = 1{\cdot}075\sqrt{\frac{p}{E}\left(\frac{d_1d_2}{d_1+d_2}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

These equations hold for two disks of the same material. If the disks are of different materials, as in the disk wear machine, then the equivalent modulus of elasticity, E_e, given by

$$E_e = \frac{2E_1E_2}{E_1+E_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

should be used.

A load commonly used for test purposes is 1000 *Sc*, where the *Sc* value is given [12] by

$$Sc = \frac{p}{R_R^{0.8}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where R_R is the relative radius of curvature, $\frac{1}{2}\left(\frac{d_1 d_2}{d_1 + d_2}\right)$

Substituting 1000 *Sc* and appropriate values for the usual test arrangement in the disk wear machine,

$$W = 408 \text{ lb}$$

Maximum compressive stress = 60,400 p.s.i. (or 27·0 tons/sq. in.)

$$B = 0{\cdot}000251 \text{ inch}$$

and

$$A = 0{\cdot}00764 \text{ inch.}$$

The order of magnitude of the value of *A* was confirmed by a static experiment on a bronze disk covered in marking-out blue.

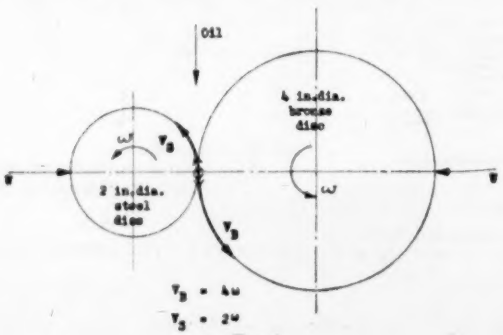


FIG 6

DIRECTION OF OIL FLOW RELATIVE TO MOVING CONTACT SURFACES OF DISKS

Lubrication Conditions

A coefficient of friction of 0·034 was obtained for the straight mineral oil A (Table I) using run-in disks on the Thornton David Brown disk machine when operated under the same conditions as the disk wear machine. The order of magnitude of this coefficient of friction suggests that the lubrication is of the mixed hydrodynamic/boundary type and not purely boundary as might be expected from the high load and opposing direction of motion of the contacting surfaces (Fig 6). The fact that the algebraic sum of the surface velocities is not zero would encourage some hydrodynamic action.[13]

WEAR

Wear during "Running-in"

The running-in process is usually considered as consisting simply of metal-to-metal contact at isolated points which continues until the major surface asperities are removed. Welds are assumed to form at the contact

of opposing high spots of the rubbing surfaces. As sliding proceeds these welds must rupture, generally plucking out metal in the immediate vicinity since the junctions will be work-hardened and tougher than the parent metal. (The presence of surface films, makes this process more complex.[14])

With the disk wear machine, when the wear rate is high, as during running-in, it is observed that the bronze disk forms yellow tracks on the steel disk (which are visible in Fig 3). This condition may be analogous to the " scuffing " encountered in steel/steel spur gears, since bronze wear-particles have evidently been transferred to the steel disk, although with the bronze disk no deterioration in its surface condition is visible and the disks continue to operate satisfactorily. The onset of the characteristic bronze discoloration appears to depend on the wear rate. For instance, tracks usually appear during running-in, when the wear rate is high, but

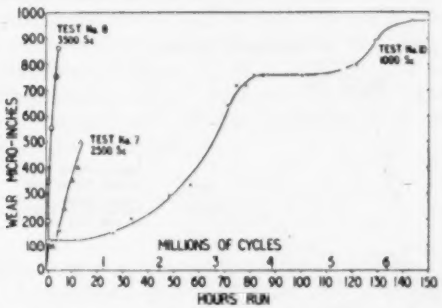


FIG 7

WEAR v TEST DURATION FOR VARIOUS LOADS

disappear later in the same test when the running-in is completed and the wear rate diminished. A " Talysurf " trace across the edge of a bronze track on the steel disk showed no detectable step due to the presence of the bronze—the track surface was merely a little rougher than the original surface.

Wear after " Running-in "

Fundamental.—When the running-in process is completed, most of the asperities will have been removed and the wear rate will decrease considerably, since the remaining asperities are generally separated from the opposing surface by the oil film. The proportion of the load carried hydrodynamically will increase and the state of lubrication will change from basically boundary to some mixed state of hydrodynamic/boundary. Running-in may be considered as ending when the asperities become small enough for the lubricant to keep the disks generally out of contact. If the lubricant is unsuitable or the load excessive, " running-in " will continue to infinity and the rate of wear will remain very high.

These effects can be appreciated from Fig 7, where the wear-versus-load curves are plotted for some preliminary tests using a single land, $\frac{3}{16}$ inch wide, previously " run-in " by five hours' operation (see Appendix III for test conditions). With a load equivalent to 3500 *Sc*, the wear rate re-

mained high until the test was concluded by the occurrence of pitting, whereas with 1000 *Sc* there was an appreciable period of operation with a very low wear rate. It is this normal low wear rate which will now be considered.

The surface layer * will be work-hardened by the repeated compressive and tensile deformation and so will become brittle. It is suggested [15] that very small particles will then be detached. (This is " wear " as defined above.) If so, then the rate of wear will be slow at first and later increase as the layer becomes more brittle. Eventually most or all of this surface layer will be removed, exposing a fresh ductile surface. The process may be repeated † on the fresh surface, which, however, will have already been work-hardened slightly by its proximity to the original layer, while remnants of the first work-hardened layer may also remain. The new layer will therefore be work-hardened more rapidly than the first

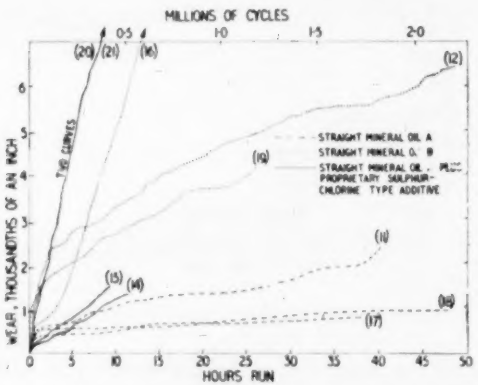


FIG 8

WEAR-RATE CHARACTERISTICS OF BRONZE DISKS WHEN USING VARIOUS LUBRICANTS

one. If the process is continued, the intervals between shedding surface layers will decrease and the separate stages merge until finally the surface will wear away simultaneously with the formation of fresh work-hardened layers below as a continuous process. The initial periodic variation in the wear rate of bronze has been noted on the disk wear machine (Test No. 10, Fig 7), as well as on other rigs [15] at Thornton.

The foregoing tentative explanation of the mechanism of wear is consistent with wear being primarily a function of the fatigue properties of the bronze, or at least of the top layers of the bronze surface. The most important properties of the lubricant are probably its viscosity and anti-frictional characteristics. The former may affect the pressure distribution and the latter the frictional force component, with the result that the deformation of the disks or gears under load will be modified and the amount of work-hardening affected.

* " Layer " is used in an arbitrary sense.

† On the disk wear machine the pitting which then occurs is often too extensive to permit further operation without undue shock load.

Effect of Lubricants—Following the preliminary tests, a short programme, designed to ascertain the repeatability of results from the rig and to indicate the possible effects of variation in viscosity and inclusion of chemical-type EP additives, was carried out. It consisted of three separate tests, involving the measurement of both the time before the onset of pitting and the wear of the bronze disk, with two straight mineral oils, A and B, of different viscosities, together with one of the mineral oils plus a proprietary sulphur–chlorine-type additive.*

The tests were run with a load giving 1500 *Sc* using three lands, each $\frac{3}{16}$ inch wide, under conditions given in Appendix III. Wear results are shown in Fig 8, zero time being after the customary running-in period of five hours. (The pitting results are discussed later.) Since the repeatability of these wear results is not so good as might be desired; the grain size of the bronze, which had meanwhile been shown [15] to influence the wear rate, was examined but found to be reasonably uniform both from disk to disk and from land to land. Fig 9 shows the typical grain-size and its uniformity from land to land.

Least wear consistently occurs with the straight mineral oil of high viscosity. A low-viscosity oil and the inclusion of a chemical-type EP agent both increase the wear rate.

Pitting

Introduction

It might be supposed that the work-hardened surface becomes so brittle that comparatively large areas are affected and particles deeper than the work-hardened layer become detached with the surface layer. If this were true, however, it would be difficult to account for the absence of particles intermediate in size between the very small wear particles and the much larger granules detached by pitting.

Lanchester's early suggestion [16] that pitting is due to the welding of the material of one body to that of the other in a small region resulting in a granule being torn away can be dismissed for the same reason and also because it is an observed fact that incipient pits can be detected visually (*e.g.*, Fig 4, Ref 5) by the presence of cracks before the pits themselves are formed.

There are at the moment two main theories purporting to account for the formation of pits, the first based on the formation of internal cracks and the second on surface cracks.

Internal-crack Theory

It has been shown by Hertz [11] that the maximum shear stress due to the loaded contact of two bodies such as gears or disks occurs below the surface. Schmitter [17] and Palmgren,[18] among others, suggested that this stress causes cracks in the interior which spread to the surface, resulting in the formation of a pit. This seems quite likely, since compressive stresses result in maximum shear stresses at about 45° to the surface. Cracks starting below the surface of case-hardened steel gears have been

* The practical objection of staining the bronze was ignored for the purpose of these research tests.

observed by Ulrich,[12] while similar observations with disks have been made by Buckingham.[20] Once a crack reaches the surface it will fill with oil and the hydraulic forces exerted by the application of load will tend to cause the formation of a pit.

It might seem from this hypothesis that pit formation should proceed independently of the wear rate, but observation at Thornton suggests that pits occur during or at the climax of a period of high rate of wear. This, however, may be merely because the surface wear removes the last connexion between the incipient pit and the parent metal (see Fig 10). Thus, once steady conditions are attained, the number of pits appearing will be proportional to the wear rate, and so more pits will be apparent after a period of operation at a high rate of wear.

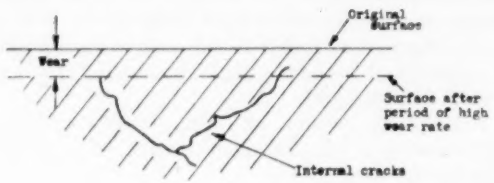


FIG 10

EFFECT OF WEAR IN HASTENING THE FORMATION OF INTERNAL-CRACK PIT

Beeching and Nicholls [21] have shown that the shear stress, S, is

$$S = \frac{2p}{\pi b^2}\left(z - \frac{z^2}{\sqrt{z^2 + b^2}}\right)$$

and that the depth of maximum shear stress is

$$z \text{ max} = 0{\cdot}786\sqrt{\frac{8p(1 - \sigma^2)R_B}{\pi E}}$$

where σ = Poisson's ratio;
b = half the minor axis of ellipse of contact; and
z = depth of stress.

The latter formula expressed in a slightly different manner is also proposed by Föppl [22] and has been used recently by Kraupner.[23]

For 1000 *Sc* conditions on the disk wear machine, assuming $\sigma = 0{\cdot}3$, the maximum shear stress is found to be $15{\cdot}5 \times 10^3$ p.s.i. (which is about the stress required to initiate fatigue cracks in the material), occurring at 0·007 inch below the surface.

Surface-crack Theory

It has been suggested by Buckingham [24] and Merritt [25] that the fluctuating compressive and tensile stresses caused by the normal and the frictional component of the load and aggravated by the high and rapidly fluctuating surface flash temperature (analysed by Blok [26]) are responsible for work-hardening of the surface and hence for the formation of cracks originating in the surface. It is also possible that minute surface cracks due to grinding

or other methods of surface finishing may already exist before the gear or disk is used. If the final grinding operation, for instance, is performed too rapidly, microscopic surface cracks may be produced.

When lubricant becomes trapped in these cracks they will be opened up by the hydraulic forces exerted, to produce pits, as are internal cracks when they reach the surface. Strong evidence for this theory comes from Way,[5] who demonstrated experimentally that with pure rolling pitting occurred only in the presence of a lubricant.

The fluctuating stresses to which the surface layer is subjected will be increased by surface irregularities, from which it follows that a good surface finish should minimize fatigue and pitting. This has been demonstrated experimentally.[5, 27, 28] The stress concentrations due to variations in surface finish are probably the main cause of the apparently haphazard distribution of the pits in any given gear or disk.

Extreme-pressure additives of the type that function by combining chemically with the metal of the surface may be expected to cause surface irregularities which will set up stress concentrations and so accelerate the onset of pitting.

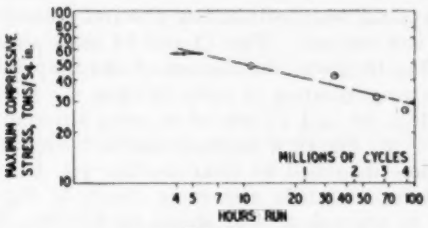


FIG 11

STRESS *v* TIME TO FAILURE BY PITTING (INCLUDING RUNNING-IN PERIOD)

Test Results

Fundamental.—Fig 11 shows the stress versus time-to-pitting results obtained from the preliminary tests * using disks with a single land, $\frac{3}{16}$ inch wide (see Appendix III for test conditions). The disk which was operated at a surface stress of 4500 *Sc* pitted during the running-in process before the start of the wear test. These results are consistent with the general trend of those of fatigue experiments, as would be expected from either the internal-crack or the surface-crack theory of pitting. Other workers [7, 29] have demonstrated the fatigue nature of pitting more extensively and shown that there is a critical stress below which pitting will not occur.

The onset of pitting appeared to coincide with a certain degree of work-hardening of the surface, hardness figures for unused disks being of the order of 115 V.P.N. and for pitted ones 145 V.P.N. The Meyer index,† another criterion of work-hardening, changes from about 2·3 for an unused disk to 2·05 at the onset of pitting.

* Plus one result from a triple-land test.

† When using a ball indentor to measure hardness, the diameter d of the impression and the load P are related by $P = ad^n$, where a is a constant and n is termed the Meyer index of the material and is proportional to its plasticity. Hence decrease in Meyer index is proportional to increase in work-hardening.

The grouping of pits in the centre of the land with an absence of pits at the edges is probably due to the metal at the edges being less restrained and therefore able to flow sideways, *e.g.*, Figs 18 and 19, so that the pressure on the surface is less.

The depths of the pits in a disk which had been operated for as long a period as possible in order to accumulate a collection of pits for study (Fig 12) were measured for comparison with the depth of maximum shear stress. (A convenient and accurate method was found to be focusing down the pits with a projection microscope.) From the results obtained from this special disk, and from other disks tested in a routine manner, the pits were found to be of two depth categories, namely, deep pits, ranging from 0·012 to 0·017 inch and shallow pits of about 0·006 inch. If the internal-crack theory is correct, there should be some tendency for the depths of pits to be roughly equal. This is in fact so, the deep pits appearing to be restricted to depths between 0·012 and 0·017 inch. Further, as would be expected, there is an appreciable degree of work-hardening about this zone. The occurrence of shallow pits can then be accounted for by the surface-crack theory.

A metallurgical examination was made of sections cut normal to the surface through typical deep and shallow pits and ground until the deepest part of each pit was reached. Figs 13 and 14 show photographs of these cross-sections. Fig 15 shows the bottom of the deep pit seen in Fig 14, taken at a higher magnification in order to show the mode of propagation of the cracks. Figs 16 and 17 are of an area adjacent to the deep pit shown in Fig 14. In Fig 16 it appears that a triangular-shaped portion is about to become detached to form another pit, but there are cracks well below this level, as can be seen more clearly in Fig 17, which covers an area slightly to the left of that shown in Fig 16. Hardness surveys were made on the two cross-sections, the small load of 1 kg being used in order to make the survey as detailed as possible. The results are shown in Figs 18 and 19, in which the figures are Vickers Pyramid Numbers.

A few tests under pure rolling conditions with and without a lubricant have been performed on the disk wear machine, using bronze and steel

TABLE I

Time before Pitting Occurred with Various Lubricants

Lubricant	Lubricant type	Viscosity, Redwood I at 140°F, sec	Time for first pit to occur (including running-in time), hours to nearest hour					
			1st test	2nd test	3rd test	4th test	Average	
							Time	No. of cycles
A	Straight mineral oil	330	60 (11)	90 (17)	67 (18)	—	72	$3{\cdot}25 \times 10^6$
B	Straight mineral oil	45	30 (12)	17 (16)	31 (19)	—	26	$1{\cdot}15 \times 10^6$
C	Oil A plus proprietary sulphur/chlorine additive	302	16 (14)	12 (15)	35 (20)	47 (21)	27	$1{\cdot}20 \times 10^6$
D	Oil A plus sulphurized fat	331	13 (24)	17 (25)	27 (26)	—	19	$0{\cdot}85 \times 10^6$

Note. Test Nos. are given in brackets to demonstrate the test order.

disks of 3 inches diameter and 1 : 1 ratio gear coupling. The results, although not conclusive, appeared to confirm Way's finding that pitting does not occur unless a lubricant is present. When the disks were lubricated, the surface hardness of the bronze disk changed in a similar manner to that of disks operated under the usual combined sliding and rolling conditions, but when they were operated dry, the hardness increased still further to 190 V.P.N. and the Meyer Index fell to 1·9. This emphasizes the part played by the lubricant in causing pits, since the surface material itself can apparently withstand further work-hardening without pit formation.

Effect of Lubricants.—As mentioned earlier, a short programme was completed on three different oils, under conditions listed in Appendix III and a load giving 1500 *Sc.* The pitting results are given in Table I, together with some later results obtained on Lubricant D, which consisted of Lubricant A plus a sulphurized fat.

The following tentative conclusions regarding the effect of the lubricant on pitting may be drawn :—

(*a*) A high-viscosity oil delays the onset of pitting more than a low-viscosity oil. This was also found by Meldahl,[8] who noticed that the lower-viscosity oil gave rise to pits of larger area. This was so in the Thornton tests.

(*b*) The addition of a chemical-type EP agent accelerates the onset of pitting.

These conclusions have since been confirmed from the field.

General Discussion

It seems probable that the deep pits may start from internal cracks and the shallow ones from surface cracks. Whether internal-crack pits or surface-crack pits appear first will depend on the conditions existing in the particular application. Although the shallow pits must cause local stress concentrations and are in fact observed to spread on continued operation, the seeming absence of pits intermediate in depth between the deep and the shallow types indicates that spread of shallow pits is not the cause of the deep pits.

Buckingham [20] mentions the occurrence of both deep and shallow pits in steel gears and considers that the shallow pits are " arrested " pits due to surface cracks. While it is probably true that arrested pitting, that is, the occurrence of non-spreading pits on a new gear generally attributed to the presence of surface irregularities prior to running in, is always due to surface cracks, it does not follow that surface cracks and shallow pits will not appear during prolonged operation after the running-in process is completed. Indeed, his experiments [24] with a phenolic laminated disk (bakelite) which darkened when overheated, showed the existence of two highly stressed regions, one at the surface and the other in a band below the surface. This suggests that pits may in general be initiated by both surface and internal cracks.

" Arrested " pitting of the bronze disks on the disk wear machine has never been observed. That is, no shallow pits have yet formed in the first few hours of operation. When they do eventually form, they tend

to increase in area and are not "arrested." It is possible that arrested pitting might be observed if the original surface was rougher and similarly arrested pitting might occur with bronze worm gears with a rough surface finish. However, with bronze as distinct from steel, it seems more likely that a rough surface will be so rapidly worn smooth that the higher stress concentrations will not persist for long enough to cause fatigue and cracks, and consequently "arrested" pitting is unlikely to occur. However, even the lower stress concentration caused by the smoother surface will eventually cause surface cracks and shallow pits, but these will be progressive.

It may well be that a rough surface finish on the steel disk or gear mating with bronze will cause increased stresses on the bronze, quicker work-hardening, and hence earlier pitting. It is, of course, also likely to cause increased wear, and may prevent completion of the running-in process.

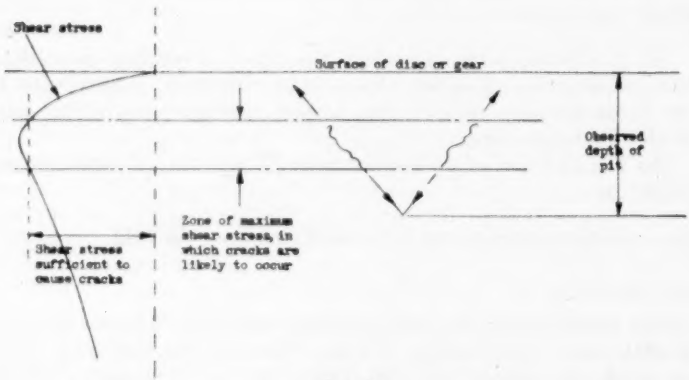

Shear stress
Surface of disc or gear
Observed depth of pit
Zone of maximum shear stress, in which cracks are likely to occur
Shear stress sufficient to cause cracks

FIG 20

FORMATION OF PIT DUE TO INTERNAL CRACKS

It is interesting to review the depths of pits observed by some previous investigators, as shown in Table II.

These results, as well as others [24] too lengthy to tabulate here, when considered collectively, support the suggestion that both deep internal-crack pits and shallow surface-crack pits occur. Kraupner [23] considered his single 0·012-inch-depth crack to be exceptional, but it may have been that his test period was too short to allow more than this one internal crack to manifest itself. The depth of internal cracks or pits seems to be consistently a little greater than the theoretical depth of maximum shear, but it must be remembered that the formula is strictly applicable only to pure rolling conditions and that the additional frictional component might be expected to increase the depth of maximum shear stress. Moreover, cracks may extend into the interior of the metal as shown in Fig 20, besides breaking out to the surface.

Because internal and surface cracks are both fatigue phenomena, the metal should be treated to give the maximum fatigue strength compatible with other requirements. The viscosity and perhaps the wetting properties

TABLE II

Depths of Pits Observed by Some Previous Investigators

Year	Source	Material	Contacting conditions	Theoretical depth of maximum shear stress (for pure rolling), inches	Depth of cracks or pits, inches		Investigator's comments	Notes
1935	Way [20]	Steel	Pure rolling	0·007	0·0008 and 0·002	—	Considered to be surface cracks	Only two isolated depths quoted
1950	Kraupner [23]	Ball-bearing steel	Sliding and rolling	0·0095	0·00016 to 0·00032	One instance of 0·012	The isolated 0·012 considered exceptional	
1951	Buckingham [20]	Case-hardened steel	Sliding and rolling	0·015	0·003 and 0·005	0·020	Considered to be surface and internal cracks	Case depth 0·020 inch
		Soft steel	Sliding and rolling	0·015 (a)		0·021 max. & 0·025 max.		Only two tests quoted
1952	Evans and Tourret	Bronze	Sliding and rolling	0·007	*ca* 0·006	0·012 to 0·017	Considered to be surface and internal cracks	
			Pure rolling	0·0085	*ca* 0·0075	—	Considered to be surface cracks	Only limited data available since pure rolling tests were performed mainly to substantiate that no pits occurred without a lubricant

Note. (a) Calculated from author's data, assuming values for E and σ.

of the lubricant will be important, since they may affect the ease with which the lubricant gains access to the crack, and so influence the tendency to pit (and probably also the size of any pits that do form).

Although these tests demonstrate that a high-viscosity lubricant is desirable, it is sometimes suggested that a low-viscosity lubricant would be preferable to reduce viscous drag and hence power losses. This point serves to emphasize that an oil for a bronze worm gear must have many attributes besides those considered here of giving low wear and long life, and sight should not be lost of these other factors when applying the results of the tests reported here. The choice of the best compromise of viscosity is complicated in practice by considerations of operating temperature and the possibility of churning in the worm-box. A heavier oil might in some applications churn more than a lighter one until its viscosity became the same as that of the lighter oil, while there have been instances of a heavier oil operating at a lower equilibrium temperature either because of a lower coefficient of friction or perhaps because a higher proportion of the load was carried hydrodynamically.

In service the wear may increase more rapidly than would be expected from results obtained from the disk-wear machine, because filtering may be less efficient or even totally lacking and detached particles may circulate within the worm-box.

Conclusions

The surface of a bronze disk operated under simulated worm-gear conditions work-hardens, suggesting that wear is caused, under this mixed hydrodynamic/boundary state of lubrication, by the surface becoming sufficiently brittle for small particles to become detached.

Pitting is initiated by either surface or internal cracks or both, according to circumstances. Surface cracks may be caused by the fluctuating compressive and tensile stresses (aggravated by the fluctuating surface flash temperature), by the method and rapidity of machining used to form the surface, or by surface irregularities. Internal cracks may be caused by a high internal shear stress, and may eventually reach the surface. Once cracks reach the surface they are propagated by the hydraulic forces exerted by any lubricant trapped in them to form pits. A lubricant of high viscosity without any chemical type EP agent would appear to be desirable for bronze worm gears to give low wear and long life before pitting occurs.

This paper is in the nature of a progress report, and further variables to investigate and tests to perform readily suggest themselves. It is hoped to continue this investigation as time permits.

Acknowledgments

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Appendix I

Metallurgical Specifications of Steel and Bronze Used

Bronze

Composition :	Copper 87·8% Tin 11·5 ± 0·5% Phosphorus 0·7 ± 0·05%
Process :	Spun cast
Hardness :	115 ± 10 V.H.N.

Steel

Specification :	En 34 (Equivalent to SAE 4615) 2% Ni–Mo
Case depth :	0·030 inch
Hardness :	800 ± 50 V.H.N.
Surface finish :	5–10 micro-inches.

Appendix II

Mechanical Tests on Bronze Used

Mechanical tests on a Hounsfield extensometer were carried out on tensile specimens manufactured from the actual disk material in use. The average results are as follows :

Ultimate tensile strength :	16 tons/sq. in.
0·1% Proof stress :	10 tons/sq. in.
Young's modulus, E :	14×10^6 p.s.i.
Elongation on 1 inch :	1½%.

Appendix III

Test Conditions

Lubricant :	Type :	Straight mineral oil
	Viscosity :	330 sec Red I at 140°F
	Flow :	2 Imp. pt. per min per land
	Temperature :	70° C
Drive :	Type :	Chain and sprocket
	Ratio :	1 to 1
Disk speed (both) :		750 r.p.m.
Sliding speed :		1180 ft/min.

Discussion

Professor W. A. Tuplin : An important factor in the lubrication of gear teeth is what has been called the " slide–roll " ratio. Inasmuch as the ordinary meanings of " rolling " and " sliding " are phenomena sharply distinct from one another, the suggestion that both may occur at once may be a little puzzling, and accordingly some alternative to the term " slide–roll " may be preferable.

By " slide–roll " ratio is meant the ratio of the sliding velocity at a line of contact to the velocity of that line of contact over the surface particularly under consideration. (In general, the slide–roll ratios of the two surfaces are different.) The velocity of the line of contact over a surface has been called the " rolling velocity "; this is appropriate if the sliding velocity is zero, but is of questionable propriety in other circumstances. It is now suggested that the velocity of the contact line over a surface be called the " sweep velocity," the ratio of sliding velocity to sweep velocity then becoming the " slide–sweep " ratio.

It is clear that if the slide–sweep ratio is greater than unity (as on 2 in Fig 1, A and B) the part of the oil film that lies on the contact line at any instant tends to be drawn over the surface in the common direction of sliding and sweeping at a velocity higher than the sweep velocity. Thus, oil that has been heated by passage through the contact zone is drawn on to the cool oil on a part of the surface that the contact line is approaching. The tendency is thus to lay a carpet of hot oil, possibly containing particles of metal detached from the mating surface, in front of the advancing contact line.

If the slide–sweep ratio exceeds 2 (Fig 1, A) even the mean velocity of the oil film exceeds the sweep velocity, and so oil may be " rolled " ahead of the contact line, and deposited on the oil film already on the surface.

If the slide–sweep ratio is 2 (Fig 1, B), the mean velocity of the oil film is equal to the sweep velocity, and there is consequently a tendency for a cylinder of oil to rotate with its axis between the nominal contact lines, while above and below it oil films are being drawn away in both directions to pre-heat the films approached by the contact lines on both surfaces. This condition is therefore exceptionally unfavourable to lubrication.

If the slide–sweep ratio is between 0 and 1 (Fig 1, C), the contact line is moving faster than the mating surface, and is therefore moving continuously on to unused oil. To this extent the conditions are favourable to lubrication, although the movement of the mating surface has a slight tendency to draw oil from the contact zone.

If the slide–sweep ratio is zero (Fig 1, D), there is no sliding and no lubrication problem.

If the slide–sweep ratio is negative (Fig 1, E), oil from the contact zone is swept over the surface in the direction opposite to that in which the contact line is moving, and there is thus no chance of pre-heating of the film towards which the contact-line is advancing. On the contrary, unused oil on the mating surface is drawn towards the contact line in a manner tending to build a hydrodynamic film there. Negative slide-sweep is thus very favourable for lubrication.

If the slide–sweep ratio on surfaces 1 and 2 are S_1 and S_2 respectively, the relation between them is

$$S_2 = \frac{S_1}{S_1 - 1} \text{ or } S_1 = \frac{S_2}{S_2 - 1}$$

The most favourable condition for lubrication is for S to be negative, but the relation between S_1 and S_2 shows that this cannot be the case on two mating surfaces at once. If S_1 is negative, then S_2 lies between 0 and 1, and this is the next most favourable condition. It happens that in spur, helical, and bevel gears the condition of negative slide–sweep on one surface and positive slide–sweep less than unity on the other is observed at every phase of engagement, and this is fortunate, as a pair of steel surfaces cannot satisfactorily carry heavy loading in any other condition. To achieve the best possible lubrication conditions, the ratio of depth of tooth to pitch-circle diameter should be made as small as possible, and the addenda of the mating gears should be equal. If this is done steel/steel gears can be satisfactorily lubricated at the highest speeds envisaged at present; at the much lower speeds of normal practice there is no need to consider lubrication as a factor affecting tooth proportions, since the slide–sweep ratio never exceeds unity unless the base circle lies within the depth of the tooth, and this is a condition to be avoided in any case.

The slide–sweep ratio is negative inside the pitch circle and positive outside it. The existence of an important difference in lubrication conditions on the two parts of the tooth-flank was first brought to the writer's notice by a pair of helical gears in which the teeth of both pinion and wheel showed a faint, polished streakiness perpendicular to the tip down to half-depth, whilst the remainder of the working flank was pitted. Outside the pitch circle there was incipient scuffing and no pitting; inside it there was pitting but no apparent tendency to scuffing. At any point of contact, therefore, one surface was tending to scuff, whilst the other was not. As the surfaces at a contact point have a common sliding velocity, there must be some other significant factor, and that appears to be the slide–sweep ratio.

On worm-wheel teeth, slide–sweep ratios greater than 2 are common and a steel worm-wheel will not work satisfactorily with a steel worm at any loading that would be economic in industry. On the other hand, a bronze worm-wheel and a case-hardened steel worm can take heavy loads at high speeds.

This contrast is confirmed by disk-machine tests; if the surfaces of the disks move in the same direction (not necessarily at the same speed) at the line of contact (negative slide–sweep on the slower-running surface) a

steel/steel combination will carry a useful load. If, on the other hand, the surfaces move in opposite directions at the line of contact (slide–sweep ratio greater than 1 on each surface) a steel/steel combination will scuff under negligible loading, whereas a bronze/steel combination may carry an appreciable load. Even in the worst case of a slide–sweep ratio of 2 on each disk, the bronze–steel combination may carry load. Overload of such a combination does not cause sudden failure, but material is torn from the bronze surface and is either carried away by the oil or adheres to the steel surface as a very thin film or in separate particles. Aggravation of this condition induces a high non-uniform coefficient of friction, and severe vibration occurs in the disk machine.

In worm-gears the combination of case-hardened steel and phosphor bronze has a high load capacity with negligible wear, despite the unfavourable slide–sweep ratios, but any considerable departure from a tin content of 12 per cent leads to inferior performance. This sensitivity to exact details of the chemistry and metallurgy of the bronze is a difficulty in attempting to explain the mechanism of lubrication in worm-gears or wherever else a slide–sweep ratio exceeds unity.

The surfaces have velocities V_1 and V_2 at the line of contact, represented here by the point of contact of the arcs.

A. Velocities in opposite directions $V_1 > V_2$
$S_2 > 2$ $1 < S_1 < 2$ Lubrication bad

B. Velocities equal and opposite
$S_2 = 2$ $S_1 = 2$ Lubrication bad

C. Velocities in same direction $V_1 < V_2$
$0 < S_2 < 1$ S_1 negative Lubrication good

D. Velocities equal and in same direction Lubrication superfluous
$S_2 = 0$ $S_1 = 0$

E. Velocities in same direction $V_1 > V_2$
S_2 negative $0 < S_1 < 1$ Lubrication good

FIG 1

DR F. T. BARWELL: In connexion with the papers by Cameron and McEwen I want to refer to the idealization represented by the replacement of gears by rotating disks. One imagines a gear tooth rotating through space as oil is squirted at it, some of which is thrown off by centrifugal force. The tooth carrying the remaining oil then suddenly comes into contact with another tooth, with or without tip relief. There is sliding and rolling,

then rolling, then sliding and rolling again all in a fraction of a second. Can we represent those conditions by a steady-state mathematical treatment? I do not say we cannot, but I think Cameron or McEwen might have helped our imagination by demonstrating the applicability of that treatment to what at first seems to be an entirely transient phenomenon.

Evans and Tourret remark that there is a fatigue limit and quote other workers as having shown that there is a critical stress below which pitting will not occur. That may be true under the conditions which those workers employ; but I think it can also be stated from experience with ball and roller bearings that, if there is a fatigue limit, it is certainly too low to have any practical significance whatsoever.

DR A. CAMERON: I am very interested to see Evans and Tourret have found a stepped wear–time curve (their Fig 7) similar to my friction–time curve.* Their explanation is most ingenious, and I wonder if it would be possible to follow the removal of these layers either by microhardness tests or by electron diffraction.

At Pametrada we have carried out a fairly full investigation of pitting on through hardened steels. We find that the cracks start from the surface and they vary uniformly in depth; we have had no trace of the two kinds mentioned by the authors. We also find that the grain flow of the surface, brought about by the skin friction, determines the direction of the crack. Thus a driven gear has cracks going away from the pitch line, and in a driving gear they are oriented towards it.

L. S. EVANS and R. TOURRET: We should like to present some additional results on depths of pits, obtained since our paper was compiled, from the roller fatigue rig.† This is a small rig in which rollers from roller-bearings are operated under sensibly pure rolling conditions in peripheral contact with three large steel rollers. Some special ½-inch-dia rollers manufactured of bronze were used, and the results are given in Table A, with columns arranged as in Table II in the paper.

TABLE A

Additional Observations on Depths of Pits

Year	Source	Material	Contacting conditions	Theoretical depth of maximum shear stress (for pure rolling), inches	Depth of pits, inches	
1952	Roller fatigue rig	Bronze	Pure rolling	0·003	0·0005 to 0·001	0·004 to 0·006

We are indebted to E. T. Hutt for the idea of an effective graphical method of presenting the results shown in our Table II. The depths of

* *J. Inst. Petrol.*, 1949, **35**, 126.

† Designed and operated by N. A. Scarlett.

pits are expressed as a ratio in terms of the depth of maximum shear stress (Table B) and then plotted (Fig 1).

TABLE B

Ratio of Actual Depths of Cracks or Pits to Depth of Maximum Shear Stress

Source		z max.	Depth of cracks or pits, inches		Depth of cracks or pits in terms of depth of maximum shear stress, inches	
Way		0·007	0·0008 and 0·002	—	0·114 and 0·286	—
Kraupner		0·0095	0·00016 to 0·00032	0·012	0·017 to 0·033	1·3
Buckingham		0·015	0·003 and 0·005	0·020	0·200 and 0·333	1·33
		0·015	—	0·021 and 0·025	—	1·40 and 1·67
Evans and Tourret	Disc wear m/c	0·007	*ca* 0·006	0·012 to 0·017	*ca* 0·858	1·72 to 2·43
		0·0085	*ca* 0·0075	—	*ca* 0·883	—
	Roller fatigue rig	0·003	0·0005 to 0·001	0·004 to 0·006	0·167 to 0·30	1·33 to 2·0

This shows at a glance the relation of the various experimental results to the theoretical depth of maximum shear stress, and may facilitate discussion.

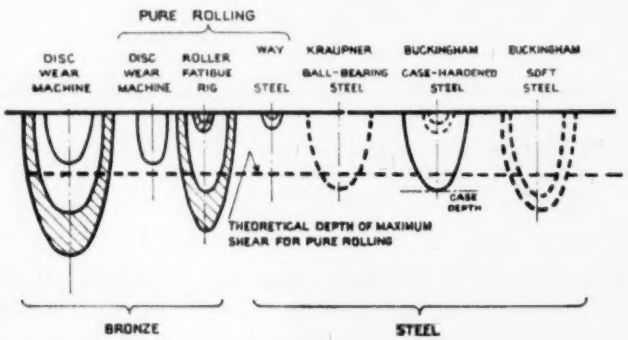

FIG 1

GRAPHICAL PRESENTATION OF DEPTHS OF PITS OBSERVED BY VARIOUS INVESTIGATORS

R. TOURRET: I should like to put in a plea that Smith might possibly give us some details of the worm gear tests that he mentions in his paper; and in particular, I should like to know his criterion of load-carrying ability.

PROFESSOR H. BLOK (written contribution): Cameron is to be congratulated on his able synopsis of the difficult subject of the theory of hydrodynamic lubrication of gears. Although the present status of the theory still leaves much to be desired, this is certainly not the fault of the author, who has contributed much to the subject. The discussion which follows is, therefore, not to be taken as a criticism, but as an attempt to indicate further lines of attack.

Under Section V the author mentions the far-reaching speculation that, when loads are high enough for the disks to become locally flat and thus render Hertz's pressure distribution applicable, minimum film thickness h_0 might become independent of load. However, on the following grounds the argument advanced is thought to be unconvincing.

It can be shown from Reynolds's differential equation for the hydrodynamic formation of pressure in oil films, that Hertz's pressure distribution cannot be exactly reproduced by hydrodynamic means because it is semi-elliptical and thus shows infinite pressure gradients at either end of the film postulated. Therefore, the discusser prefers the following more elaborate starting point, which led him a few years ago to a speculation identical with that in the paper.

With the proviso that under the high loads contemplated pressure formation is still fully hydrodynamic—whilst it is, of course, appreciably affected by the elastic deformations of the rubbing surfaces—it can be shown, from the theoretical considerations of Peppler,* that the distribution of hydrodynamic pressures must approach very closely to Hertz's distribution in about the way indicated in the model † of Fig 2 (*cf* the near-Hertz with the Hertz distribution).

As regards the maximum elasto-hydrodynamic or say, near-Hertz pressure, this must be nearly equal to Hertz's maximum pressure. This is true, irrespective of whether the tendency of viscosity to increase under the high pressures is or is not appreciably counteracted by its tendency to decrease under the temperature rise evoked by the viscous-frictional heat. A corollary of this finding is that there can be no question of the occurrence of critical cases, such as treated in Section III, where maximum pressure is considered to become infinitely high (see also the paper by McEwen).

Further, the near-Hertz pressure formation is characterized by the feature that it is essentially confined to an "active portion" of the oil film, the length of this film portion being about equal to the width L of the flat contact area that would be produced under the Hertz conditions

* W. Peppler: "Untersuchung über die Druckübertragung bei belasteten und geschmierten umlaufenden achsparallelen Zylindern"; Maschinenelemente-Tagung Aachen 1935, 42–43; V.D.I. Verlag, Berlin, 1936 (see also VDI Forschungsheft No. 391, 1938).

† A simplifying feature of the model is, that the elastic deformations of both surfaces have been concentrated into one, *i.e.*, the upper surface; accordingly, the other, lower surface has been assumed to be perfectly rigid, so that it retains its plane contour. This simplification can be shown to be entirely permissible.

if they were applicable. This feature ensures that for any given load the near-Hertz distribution of elasto-hydrodynamic pressures as compared with Hertz's distribution of contact pressures shows an asymmetry, in that there is a pressure surplus at the inlet, and a pressure deficit at the outlet of the film; this asymmetry introduces the tilting action necessary for making the film convergent and thus hydrodynamically effective. A related asymmetry consists in the feature that the active portion of the film is shifted somewhat towards the film inlet. These two asymmetries render the way in which the hydrodynamic pressures are affected by the shape of the film compatible with the way in which the shape of the film is in its turn affected by the elastic deformations induced by the hydrodynamic pressures. Thus, the active portion of near-Hertz films may assume a shape as indicated (see Fig 2) which is curved only slightly, so that it is very similar to the straight-wedged films in Michell thrust bearings.

For such near-Hertz films, and as long as viscosity is considered uniform throughout the film, minimum film thickness h_0 is, therefore, related to the her variables by :

$$h_0/L = H\sqrt{\eta V/W}$$

where the factor H represents a function of the ratio $h_1/h_0 \gg 1$, in which h_1 stands for film thickness at the inlet of the active portion of the film.* Now, in the author's proportionality $h_0 \propto \sqrt{\eta V/W}$ (see Section V), the " constant " of proportionality, H, has been omitted. This may only be done, however, if H can be proved to be really constant, *e.g.*, independent of load W. Admittedly, it can be shown that the above-mentioned tilting action automatically adjusts itself to increasing load in such a way that the ratio h_1/h_0 diminishes only slowly. In other words, the factor H may be expected to remain approximately constant over each of a number of successive, more or less arbitrary, ranges of near-Hertz cases (which ranges might possibly be chosen fairly wide). Nevertheless, in actually reaching Hertz's limiting case, minimum film thickness would vanish, H becoming successively (properly continously) smaller.

It is remarkable that in extrapolating an earlier approximative elasto-hydrodynamic theory—which has previously been outlined †—towards the near-Hertz cases, a constant value ensues for H, viz., of the order of unity.‡ However, this extrapolation is not really justified, because the approximations characterizing the relevant theory had been adapted especially to the other extreme, *i.e.*, where elastic deformations are comparatively slight. For this reason the speculation concerning the constancy of H was not reported at the time. In the meantime another approximate elasto-hydrodynamic theory has been evolved, which is based

* More precisely, H is a function of the entire shape of the active portion, but, on account of the latter's slight curvature H will be sufficiently defined by only one shape parameter, h_1/h_0.

† See Fig 7 and the corresponding text of the discusser's paper; " Fundamental mechanical aspects of thin-film lubrication," *Annal. N. York Ac. Sci.* 1951, **53** (4), 779–804.

‡ According to this particular approximate theory, the elastic deformations would so adjust themselves to load, that the radius of curvature R of the rubbing surfaces varies in the same manner as minimum film thickness h_0. Accordingly, the ratio h_0/R would remain constant, so that coefficients of friction (*cf* equation (IX)) would also remain constant.

on the model of Fig 2, and thus is much better adapted to the near-Hertz cases.

The numerical results so far available tend to show that in fact H is not constant, although it seems that the variation of H is slow. The calculations are being pursued and verified, because from a design standpoint it is very important to answer the question whether there is a range of near-Hertz cases in which minimum film thickness is only slightly affected by load, and if so, in what circumstances. If loads at the commencement of such a range would result in a permissible minimum film thickness, they could be increased further, possibly much further, whilst maintaining safe hydrodynamic lubrication.

In view of the importance of this question about the potentialities of the hydrodynamic lubrication of gear teeth, it seems justified to supplement the above, rather vague, indications by data on an *upper limit* which can be assigned to the minimum thickness in near-Hertz cases. Such data may often suffice for judging whether there is perhaps a possibility of achieving a permissible minimum film thickness; this thickness should not be smaller than the height of the irregularities on the tooth faces.

In deducing the upper limit one can conveniently start from the result of Lord Rayleigh * for the optimum film thickness h_{opt}, conceivable for a film of given length L, viz.,

$$h_{opt}/L = 0{\cdot}454\ \sqrt{\eta_1 V/W}$$

Substituting L from Hertz's well-known formula (for R see the author's definition (IV)), *i.e.*,

$$L = \sqrt{32/\pi}\,.\,R\,\sqrt{W/ER}$$

where E represents the " reduced " modulus of elasticity,† it follows from $h_0 \ll h_{opt}$, that,

$$h_0/R \ll 1{\cdot}45\ \sqrt{\eta_1 V/ER}$$

Because this upper limit has been deduced under the assumption of a viscosity uniform throughout the film, it might be argued that the enormous local increases that viscosity undergoes under pressures of the Hertzian order of magnitude, might bring about a substantial increase of the optimum film thickness. However, it can be shown that the factor 1·45 could only be doubled or perhaps tripled. Even so, this comparatively slight beneficial effect of pressure viscosity may well be offset by the attendant viscous-frictional temperature rise in the oil film postulated.

Further, it might be argued that temperature rise of the oil, reckoned from the inlet towards the outlet of the film, might become so great that Fogg's thermal-wedge effect would greatly enhance hydrodynamic load-carrying capacity. However, the relevant beneficial effect of the increase of specific volume of the oil by thermal expansion can easily be offset by the compression of the oil under the high film pressures postulated; *e.g.*, with mineral oils a temperature difference along the film of at least 200° C is

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* Lord Rayleigh : " Notes on the theory of lubrication," *Phil. Mag.*, **35** (6), 1–12. Note that the factor 0·454 in the above formula is the square root of Lord Rayleigh's factor 0·2063.

† E is defined by $1/E = \frac{1}{2}[(1 - \nu_1^2)/E_1 + (1 - \nu_2^2)/E_2]$, where E_1 and E_2 are the moduli of elasticity, and ν_1 and ν_2 Poisson's ratio for the two mating surfaces 1 and 2.

needed for overcoming the compression that these oils undergo under a pressure of 10,000 atmospheres.

As regards the conceivably beneficial effects of the variation of temperature, and thus of viscosity, *across* the film (see Section IV), a recent paper * may be referred to. There it was indicated that in any film section, where the film is really thin, the viscous-frictional heat developed in that section is so effectively conducted away across the film towards the rubbing surfaces that the effects of convection of the heat by the oil flowing towards the next sections becomes negligible. Moreover, the cross-conduction of the heat would appear to be so effective, that no appreciable variation of temperature would occur across the film.† Therefore, it is not thought that viscosity variation across the film can contribute much to the hydrodynamical load-carrying capacity of spur gears.

Further, it might be argued that the inertia effects to which the oil in the film is subjected by the variations in the tangential velocities of the meshing tooth faces relatively to their area of closest approach may perhaps promote the hydrodynamic formation of film pressure and thus of hydrodynamic load-carrying capacity. It should then be taken into account, however, that whereas one of the two faces will show a tangential deceleration, the other will show a tangential acceleration. From the present point of view, the deceleration has indeed a beneficial effect, but on the other hand, the acceleration has an adverse effect.‡ In the particular case that both mating gears have the same geometry, the two effects may be expected to cancel. Therefore, in spur-gear rigs of the IAE type, in which the two test gears have about the same geometry (gear ratio being about equal to unity), the inertia effects are considered to be rather insignificant.

Finally, it might be argued that permissible minimum film thickness may be less than the height of the surface irregularities, as ordinarily measured, *i.e.*, in the unloaded condition of the tooth faces. In fact, the irregularities will flatten elastically, *i.e.*, intermittently, each time they pass through the high-pressure region of the oil film postulated. But it is not thought that these micro-deformations could be so pronounced as to allow for permissible minimum film thicknesses much smaller than the one indicated by the ordinary measurements of roughness.

All in all, unless some hitherto unknown beneficial effect, which would have to be powerful indeed, has been overlooked, it would appear that the order of magnitude indicated above for the optimum minimum film thickness conceivable should be about right. Under the favourable assumption that such an optimum can actually be approached in near-Hertz cases, but provided that tooth loads approach those at which scuffing occurs with straight mineral oils, even in well-finished and well-run-in gears the height of the surface irregularities would exceed the optimum minimum film thick-

* See the reply to discussion on "Viscosity–temperature–pressure relationships, their correlation and significance for lubrication," *Proc. 3rd World Petr. Cong.*, 1951, **VII**, 304–19.

† Accordingly, flash-temperature theory, in a form adapted to hydrodynamic lubrication, could be applied here.

‡ For journal bearings, in which the shaft has a non-uniform speed (*e.g.*, during starting and stopping), this has been demonstrated theoretically by D. J. Ladanyi in his paper, "Effects of temporal tangential bearing acceleration in performance characteristics of slider and journal bearing," *Nat. Adv. Comm. Aeron. Techn. Note No.* 1730, Washington D.C., 1948.

ness to such an extent, that fully or essentially hydrodynamic lubrication is not conceivable in the light of present knowledge.

In conclusion, the discusser heartily agrees with the author that much more experimentation on the potentialities of hydrodynamic lubrication is needed before the above conclusions can be confirmed or refuted in a really convincing manner. As far as the experimental study of the important elasto-hydrodynamic effects are concerned, it is worth while mentioning that rubber or other highly deformable models of gear teeth or disks have shown to be promising, as they render several of the measurements needed much more practicable than is the case for the comparatively rigid surfaces of metal.

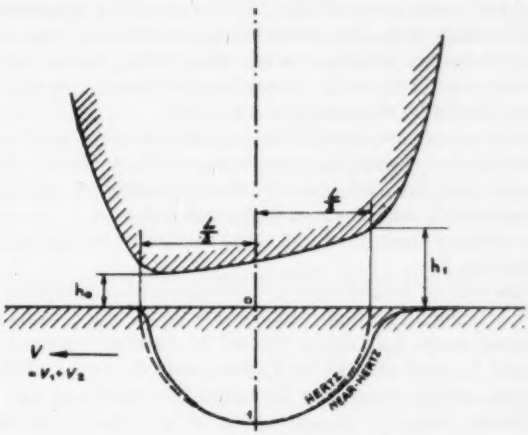

FIG 2

NEAR-HERTZ MODEL

In their para (*c*) Finch and Spurr have implied that two opposing rubbing surfaces need not reach the same temperature; more specifically, the surface, along which an irregularity rubs in a long contact path, has been supposed not to reach such high temperatures as the other, mating surface (say, the irregularity) on which the contact path is short or even nil.

The discusser takes it that this supposition was, as suggested by the first author in at least one of his previous papers, deduced from the principle of equipartition of energy; in this deduction the principle has evidently been interpreted as meaning that the frictional heat should be divided equally between the two rubbing surfaces, that is, between the two rubbing bodies.

However, as far as the discusser is aware, it is not safe indiscriminately to apply the principle outside (either above or below) the dimensional range for which it was originally developed, that is outside the rather limited range of molecules. More specifically, it is considered unsafe, if not impermissible, to apply the principle in any direct way, such as the authors', to rubbing bodies in bulk, their usual size being very far above the molecular range.

Could the authors indicate in what manner they think their extrapolation of the principle concerned from its normal range to the dimensions of

rubbing bodies in bulk could be justified? In such a justification it would be indispensable indeed to point out the limitations of the extrapolation.

That several of such limitations must exist, follows from the extensive experimental evidence available on rubbing bodies. For instance, it has been known for decades, that in the process of cutting metals at speeds such as occur in a lathe the chip takes by far the greater part of the frictional heat generated between it and the tool; there are many more instances where the principle of equipartition of energy proves inapplicable outside its proper range.*

In regard to the paper by Barwell and Milne the discusser wonders whether the contact areas, to which the results in Tables I and II relate, are band-shaped or at least oblong. If they are, the temperature criterion $t \propto \mu p v^{\frac{1}{2}}$ applies only with the proviso that the width of the contact area is independent of load; however, when this width varies with the square root of the load p per unit width (according to Hertz's theory), the temperature criterion should be changed into $t \propto \mu p^{\frac{1}{2}} v^{\frac{1}{2}}$.

If the contact areas are more or less square, or circular, the temperature criterion should be chosen in accordance with a theory that has been developed specially for such cases,† where, moreover, one of the rubbing surfaces is stationary with respect to the contact area.

Could the authors explain why it is the first criterion that should apply to their experiments?

Finally, the author would issue a warning against the indiscriminate use of Almen's "scoring" (scuffing) criteria PV and PVT; to be more specific, the restrictions made by Almen should be carefully considered, *i.e.*, PV was developed for and should be applied only to automobile spiral-bevel rear-axle gears, which ordinarily do not differ widely in size or type, and PVT was meant only for spur gears in aircraft-propeller reductions. If such restrictions are neglected, one can arrive at quite absurd conclusions; for instance, if it is intended to deduce scaling-up model rules for the power N that is transmissible in view of scuffing, when the models are to be geometrically similar, and of the same gear material lubricated with the same oil, but of different scales. If the scale is represented by some characteristic dimension, l, and the rotational speed by n, it is readily verified that the PV criterion would lead to the model rule $N \propto n^{-1} l^0$ (the exponent zero denotes that power transmissible would be altogether independent of dimension); further, an equally unjustified extrapolation of the PVT criterion would result in the model rule $N \propto n^{-1} l^{-1}$, which is also contradictory to experience in practice.

On the other hand, the discusser's scuffing criterion (quoted in the paper) would appear to be more universally applicable in that it results in the model rule $N \propto \mu^{-4/3} n^{1/3} l^{5/3}$, where μ denotes coefficient of tooth friction ‡).

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* *Cf*, among others, F. Charron, "Partage de la chaleur entre deux corps frottants," *Publ. Sci. Techn. Min. Air (Paris)*, No. 139, 1939.

† See, for instance, "Theoretical study of temperature rise at surfaces of actual contact under oiliness lubricating conditions," *Proc. Gen. Disc. Lubrication*, Inst. Mech. Engr, London, 1937, **2**, 222–35.

‡ *Cf*, "Getriebeschmierstoff—Ein Getriebebaustoff," Schriftenreihe Antriebstechnik, Heft 1, Fachtagung Zahnradforschung, 1950, 153–83; F. Vieweg, Brunswick, Germany (for an English translation, see "Gear lubricant—a constructional gear material," *De Ingenieur*, 1951, **63** (39), pp. O.53–O.64).

In present designs of marine-gear reductions horsepower rating is chosen according to rules that amount to a proportionality of about $l^{5/2}$ to l^3 (these rules are based on the need for avoiding the risk of tooth fracture and pitting); in fact, there are indications that increasing the size of a given type of gear, whilst choosing the power rating according to the latter rules with their greater exponents, leads to the risk of scuffing as long as the same lubricating oil is used.

Concerning McEwen's paper, the author's numerical results have been compiled in Table I, which, for three values of the exponent n (6, 8, and 10), gives B as a function of the parameter ξ. Now, the quantity B can easily be visualized, because it represents the relative gain in minimum film thickness that is afforded by the hydrodynamic effects of pressure viscosity; in fact, B is the ratio between the minimum film thickness h_0 according to the author's pressure–viscosity theory and the minimum film thickness h_M according to Martin's theory * for constant viscosity (both theories relate to Reynolds's boundary conditions for pressure). However, the parameter ξ still contains h_0 (see definition (14)), which in practice is not known beforehand. It would appear more convenient to express B in terms of another parameter which contains only those quantities that are given at the outset, *i.e.*, P, μ_0, v, r, n, and k. This can be achieved in the following way.

The parameter ξ can be rewritten (*cf* definition (14)) as :

$$\xi = \tfrac{3}{4}\pi\sqrt{2} \,.\, (4 - 3\sec^2\alpha)(n-1) \,.\, (\mu_0 v/kr)(r/h_M)^{3/2}(h_M/h_0)^{3/2} \quad (29)$$

where :

$$r/h_M = \tfrac{1}{3}\sec^2\alpha \,.\, (P/\mu_0 v) \quad . \quad . \quad . \quad . \quad . \quad (30)$$

It should be noted that in the formula preceding formula (24), h_0 is appropriately replaced by h_M; further, $2{\cdot}448 \equiv 3\cos^2\alpha$.

Accordingly,

$$\xi = (\pi/\sqrt{24}) \,.\, \sec^3\alpha \,.\, (4 - 3\sec^2\alpha) \,.\, \{(n-1)P/kr\} \,.\, (P/\mu_0 v)^{\frac{1}{2}} \,.\, B^{-3/2} \quad (31)$$

so that, introducing the new parameter aimed at, N, as defined by :—

$$N \equiv \{(n-1)P/kr\} \,.\, (P/\mu_0 v)^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad (32)$$

this can be expressed in terms of ξ and B, viz :—

$$N = (\sqrt{24}/\pi) \,.\, \cos^3\alpha \,.\, (4 - 3\sec^2\alpha)^{-1} \,.\, \xi B^{3/2} = 3{\cdot}560\,\xi B^{3/2} \quad (33)$$

By means of Table I, B can now, for each value of n, be calculated as a function of the new parameter N only. For $n = 6$, this function has been plotted as the uppermost curve in Fig 3; N has been termed the hydrodynamic pressure–viscosity parameter.

The uppermost curve concerned $(n - 6)$ ends abruptly at a so-called terminal point, *i.e.*, at the critical value of N, 18·72, with $B = 2{\cdot}826$; this value corresponds to the critical value of ξ, unity, because this results in $N = 3{\cdot}941\ B^{3/2} = 3{\cdot}941 \,.\, (2{\cdot}826)^{3/2} = 18{\cdot}72$ (*cf* Table I).

It can be seen from the figure that B is roughly a linear function of N.

The lower curve in Fig 3 refers to a pressure–viscosity relationship different from the author's, in that it is exponential, viz. :—

$$\mu = \mu_0 \,.\, \exp(\alpha p) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

* See "The lubrication of gear teeth," *Engineering*, 1916, **102**, 119–21.

where α represents a constant-pressure coefficient of viscosity. This curve represents the results * obtained by the discusser (also with Reynolds's boundary conditions for film pressure) shortly after his discussion on Hersey and Lowdenslager's paper. B is here plotted against a parameter N_{exp}, defined by :—

$$N_{exp} \equiv (\alpha P/r) \,.\, (P/\mu_0 v)^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad (35)$$

which takes the place of N as defined in (32), and which has been plotted to the same scale; accordingly, α takes the place of the quantity $(n-1)/k$ in (32). The critical value of N_{exp}, at which the uppermost curve has its terminal point at a relative gain in minimum film thickness of 2·313,

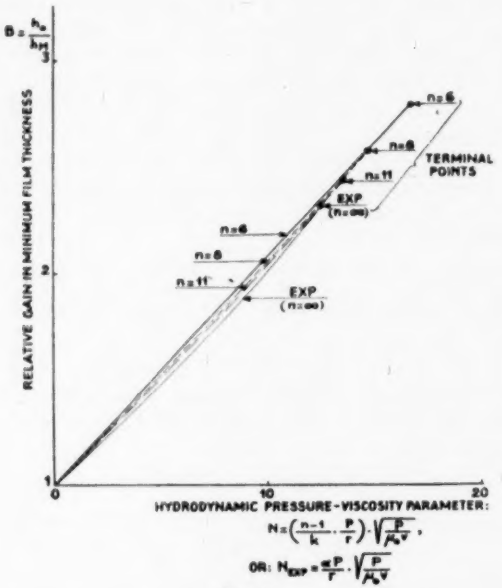

FIG 3

RELATIVE GAIN IN MINIMUM FILM THICKNESS AS A FUNCTION OF THE HYDRODYNAMIC PRESSURE–VISCOSITY PARAMETER

amounts to 12·52, which is substantially lower than the critical values of N, viz., 18·72. Nevertheless, in the region 0 to 12·52, which is common to both curves, these show pretty much the same trend.†

Besides the relative gain in minimum film thickness, the relative increase in maximum film pressure is another useful criterion for the significance of pressure viscosity; the latter criterion is defined as the ratio P_{max}/PM

* Shortly after the reading of Hersey and Lowdenslager's paper, these results were communicated privately to M. D. Hersey, in order to supplement the results that the discusser had previously obtained with Sommerfeld's boundary conditions and that he had already mentioned in the discussion on the paper concerned.

† This feature had been anticipated in "Viscosity–temperature–pressure relationships, their correlation and significance for lubrication," *Proc. 3rd World Petr. Cong.*, 1951, **VII**, 304–19.

between P_{max}, the maximum film pressure that occurs when pressure viscosity is effective, and P_M, the maximum film pressure according to Martin's constant-viscosity theory. Considering that

$$P_M = (\pi\sqrt{24}).\sec^3\alpha\,.\,(4 - 3\sec^2\alpha)(P/r)\,.\,(P/\mu_0 v)^{\frac{1}{2}} \quad . \quad (36)$$

it is easily deduced from formula (15) with $\phi = \alpha$, that with the author's pressure–viscosity relationship (8),

$$P_{max}/P_M = (n - 1)\,.\,[(1 - \xi)^{-1/(n-1)} - 1]\,.\,\xi^{-1}\,.\,B^{-3/2} \quad . \quad (37)$$

Again, from Table I P_{max}/P_M can be calculated as a function of N; for $n = 6$ this function has been plotted in Fig 4.

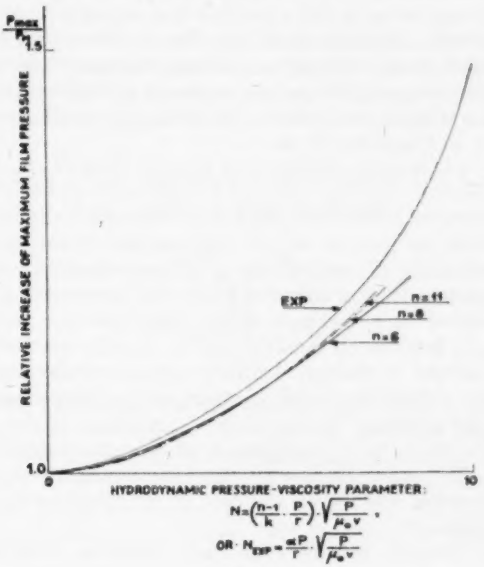

FIG 4

RELATIVE INCREASE IN MAXIMUM FILM PRESSURE AS A FUNCTION OF THE HYDRODYNAMIC PRESSURE–VISCOSITY PARAMETER

It is seen that the ratio P_{max}/P_M rises more sharply, the more closely the critical value of $N = 18{\cdot}72$ (for $n = 6$) is approached; at this critical value the ratio becomes infinite.

For purposes of comparison the relative increase in maximum film pressure has also been plotted for the case of the exponential pressure–viscosity relationship (see the uppermost curve in Fig 4; the scale on the abscissa now refers to N_{exp}, as defined by (35)). The two curves show a striking similarity in their trends, and at the lower values of the respective hydrodynamic pressure–viscosity parameters they even coincide more or less.

From Fig 3, which depicts the trend of the relative gain, B, in minimum film thickness it can be seen that when n increases from $n = 6$ to $n = 8$, and further to $n = 11$, the terminal point of the relevant curves (see the

encircled points) approaches that of the curve (marked Exp.) for the exponential viscosity–pressure relationship (34). In fact, the critical values of N are 16·70, 14·78, and 13·56 respectively (see formula (33) and Table I), and these are to be compared with the critical value of N_{exp}, 12·52, for the exponential relationship.

Furthermore, it is seen from Fig 3 that for the n-values investigated and within the range $0 \leqslant N \leqslant 12{\cdot}52$, all of the four curves show a remarkably similar trend. In other words, within this range of N and as long as $n \geqslant 6$, there are only comparatively slight differences between the author's results with the power relationship (8) and the discusser's results with the exponential relationship (34). More precisely, the higher the n-value, the closer the relevant n-curve will approach the exp-curve; there are reasons to believe that when n tends to infinity, the n-curves tend to approach the exp-curve in such a way that in the limiting case they will coincide with it.

For the whole range $n \geqslant 6$ one may even set up approximate formulas for the N_{crit}-values (that is the values of N following the formula (33) for the critical cases $\xi = 1$) and for B, viz. :—

$$B - 1 \simeq 0{\cdot}185N \simeq 2{\cdot}32\,[1 + 8{\cdot}3(n-1)^{-2}]\,.\,N/N_{crit}\,. \quad . \quad (38)$$

in which

$$N_{crit} \simeq 12{\cdot}52\,[1 + 8{\cdot}3(n-1)^{-2}] \quad . \quad . \quad . \quad . \quad (39)$$

Thus, at least as long as $n \geqslant 6$, the results of the power viscosity–pressure relationship (8) and those of the exponential relationship (34), which is simpler in that it contains only one parameter, α, may approximately be treated on a common basis; this does not only apply to the results of Fig 3, but also to those of Fig 4. Really worthwhile differences between the results of the two viscosity–pressure relationships can occur only when n is sufficiently small (perhaps appreciably smaller than 6, the lowest value investigated by the author); n-values of the order of 4 to 3 would perhaps be more representative of the peculiarities of the power relationship. It remains to be seen, however, whether such small values of n really occur with lubricating oils and, if so, whether they are of really practical importance.

Evans and Tourret remark that the additional frictional component might be expected to increase the depth of maximum shear stress. This is in contrast with what one would expect on the basis of the theoretical results by Karas * and by Poritsky,† which account for the effects of friction in the contact area on the field of stress. Could the authors state the reasons underlying their expectation?

Further, it should be noted that, curiously enough, present interpretations of the stress theories concerned do not attempt to explain why, with the band-shaped contact area that is ordinarily considered, the pits (in top view) are not band-shaped too, but more or less circular. After all, the explanation is quite simple, if one also considers the shear stresses that act in planes, parallel to the plane through the longitudinal axis of, and per-

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* F. Karas, "Die äussere Reibung beim Walzendruck," *VDI-Forschung*, 1941, **12**, 266–74 (see also the papers by the same author in: *VDI-Forschung*, 1940, **11**, 108; *VDI-Forschungsheft No.* 406, 1941; *VDI-Zeitschr.*, 1941, **85**, 341–4).

† H. Poritsky, "Stresses and deflections of cylindrical bodies in contact with application to contact of gears and of locomotive wheels," *J. appl. Mech.*, 1950, **17**, 191–201.

pendicular to, the contact area. One then finds that the maximum shear stress in such planes occurs at approximately the same depth as the maximum shear stress in the planes ordinarily considered, *i.e.*, in those perpendicular to the longitudinal axis of the contact area; it is equally important that there is not much difference either between the magnitudes of the two maximum shear stresses considered. Accordingly, the preferential directions for the cracks spreading from some weak spot at about the dangerous depth are not confined to the plane perpendicular to the longitudinal axis of the contact area, but they radiate approximately as the generatrices of a cone having its apex in the spot concerned. Such an arrangement of cracks will in fact result in the formation of more or less circular pits.

After all, it should not be overlooked that even before the stage at which cracks and flow of the material start, the theory of elasticity as used in the work by Karas and others is no longer entirely reliable; it should preferably be replaced by the theory of plasticity, or of elasto-plasticity.

PROFESSOR G. I. FINCH: I want to consider some of the remarks made by Professor Tuplin on the effect of slide/roll ratios. I should be inclined to give quite different reasons for the effects of which he spoke.

FIG 5

Let us consider two disks rotating against one another, one rotating in one direction, the other rotating in the opposite direction at similar speeds (Fig 5). Wear is then due only to compressive rolling stress effects.

Now consider one disk rotating whilst the other remains stationary, so that the contact line on one disk remains stationary, whereas the contact line on the other is always travelling (Fig 6). Heat is developed at the

FIG 6

region of contact, and is shared equally between the two surfaces. But the heat is concentrated on the fixed surface line, whereas on the rotating surface it is spread over the whole mass of metal around the periphery. The metal of one disk will wear away very rapidly. If both disks are rotating it is the periphery of the more slowly moving disk which will wear most rapidly.

Suppose there is a combination having one wheel of steel, or some such high-melting material, and the other of bronze, *i.e.*, of a much lower-melting metal. Where both are running together, in the same direction and at similar speeds, all is well, because the line of contact is shifted in both cases equally quickly around the peripheries of the two wheels. But if the steel

wheel is rotated and the bronze one kept stationary, the bronze will locally reach a very high temperature, whereas in the steel wheel the same amount of heat is distributed over the metal around the whole periphery; there will be excessive damage to the bronze, which will soon bring the thing to a standstill. On the other hand, if the bronze wheel is rotating and the steel wheel is stationary, the heat is dissipated all round the bronze wheel, and far greater speeds and pressures can be obtained between the two wheels than if the bronze wheel were stationary and the steel wheel running; the much more refractory steel can stand the high temperature much better than the bronze.

The difficulty of running steel against steel with high slide/roll ratios I would suggest is due in both cases to the high temperatures that can be reached, and which will have the same effect on both contacting surfaces. The oxide film on both surfaces will be robbed of its support, leaving bare metal, and immediately pick-up, seizure, and such-like effects occur.

But with steel against bronze, as the temperature rises the bronze protective oxide film—mainly cuprous oxide—very soon begins to soften and the metal underneath also softens, so that the oxide is robbed of its support and the bronze is laid bare. But the temperature at which that occurs is very much below that at which the iron oxide, melting at some-

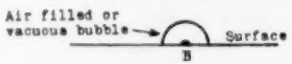

FIG 7

thing like 1450° C, is softened, and far below the temperature at which steel softens (say 1400° C). Bronze on steel, therefore, gives a far better combination for running under high slide/roll ratio conditions than does steel on steel.

I will not discuss the lower ratios, because it can be seen how the position alters. The great thing then is to aim at rigidity of the bearing-metal surfaces which have to support the essential oxide films without which the surfaces will not slide effectively.

I will now turn to one or two points which Clayton has adumbrated. He is wedded to hydrodynamic lubrication, which he quite rightly regards as the ideal, though it cannot always be attained under the highly stressed conditions nowadays. But when Clayton says it is perfect I must join issue with him. One of the examples of hydrodynamic lubrication to which I would call your attention is the propeller working in the sea. That is a case of pure hydrodynamic lubrication; yet there is the phenomenon of cavitation, causing severe erosion at the trailing edges.

I cannot see how some degree of cavitation erosion, with its possibly very serious effects, can always be avoided in gears. Such cavitation damage is sometimes attributed to fatigue or even chemical corrosion. Suppose there is a surface and on it oil, and there is a bubble in that oil (Fig 7). A sudden increase in pressure (due, say, to the rapid approach of an opposing surface) collapses the bubble towards *B*. If the bubble was vacuous a highly concentrated hammer-blow occurs at *B*. If gas filled, the bubble will also collapse, again towards *B*. There is very high adiabatic heating

of the gas contents, with the result that a tremendously high-temperature spot is generated in the surface of the metal, and gas is driven towards *B*. That is how pitting is formed on the trailing edge of a ship's propeller when bubbles form. The same thing can be envisaged happening in a gear tooth, and when it does happen there is a little pit and also the beginning of the formation of cracks.

Reference has been made to the condition of the surface in relation to its effect on the oil film. When two ideally smooth surfaces are loaded together with a thick oil film in between, the oil, at first, is easily squeezed out. As the oil film becomes thinner, however, and approaches a thickness of only one or two molecules, it is no longer a liquid but a solid. There is then an adsorbed film in which the oil-film molecules are regularly orientated by the usual Van de Waal's forces; no amount of normal pressure can express this final oil film. But it can be rubbed away by sliding.

Some claims have been made that it is advantageous to have rough surfaces, the idea being that the valleys in the surface will serve as reservoirs for the oil. On the other hand, the points of contact will be robbed of oil even more easily than will be an ideally smooth surface, and these points of contact cannot readily be supplied with oil from the valleys. Beyond capillary forces there is nothing tending to spread the oil, and the conditions of pick-up approach much more rapidly. I advocate the highest possible finish on all surfaces which have to meet rapidly and carry heavy loads, as they do in the conditions of gear operation, provided their geometry is correspondingly accurate.

Regarding graphite, I have previously pointed out the importance of using this as a lubricant only *after* it has been sheared. It is no use expecting a machine to shear the graphite flakes apart, because it can do that only by getting the flakes on edge and crushing them apart, and when on edge the graphite flakes are abrasive. If the graphite in an oil is to perform its function satisfactorily, it must already have reached that state of comminution which it will retain throughout its working life.

I do not think any of us have the right to be dogmatic about scuffing. There is still a great deal to be learned, and I will reserve some of my own ideas on that until I have more information to guide me.

To come now to the question of acidity in oil. I do not like acidity in oil. It is true that in an acid oil there is easy soap formation, and soap is a good lubricant under certain conditions. But one very harmful thing about such formation is that it takes place at the expense of the oxide layer. There is interaction between acid and oxide; the oxide layer is easily removed when combined with soap, and unless it can be replenished immediately, the oxide layer which is so vital to the successful sliding and rolling of surfaces is weakened.

I think the best answer to Blok is to describe some simple experiments which he can easily repeat. In the case of the tool against a billet in the lathe, we all know that the chip can get red hot (due to deformation work) while the tool can remain much cooler. But if he should now run the tool with the top rake so negative that the chip is no longer removed and the surface of the billet is only burnished, he will find that, although the billet remains cool, the tool tip fuses handsomely. Again, let him take a disk of Wood's alloy (m.p. 68° C) and run against this a camphor tip;

he will find that the camphor melts and polishes whilst the Wood's alloy shows no sign of having melted. Or run a 1-cm steel disk against a 15-cm brass disk rotating at 1500 r.p.m.; the small disk will carry a load of 15 or 20 lb without seizure if lubricated. But reverse the arrangement and have a small brass disk sliding against a larger steel disk, and seizure will rapidly occur even with loads of less than 1 lb. To quote another instance, run cotton thread rapidly through a hard steel eye, and observe how the cotton cuts through the eye. There are other examples, but I think that these will suffice.

PROFESSOR MCEWEN: I do not want to defend my paper now, and I hope I shall not be accused of giving tit-for-tat with Barwell; but I should like to comment on this question of a pure sliding test in connexion with gear problems. Tuplin has emphasized that in a gear problem we never have to deal with pure sliding; and although one can criticize the use of disks as simulating gear-teeth conditions, I think we can criticize pure sliding even more so.

Evans and Tourret are sure that there is a fatigue limit in pitting; I do not think there is. I have seen cases in which pitting has occurred after 6×10^{10} cycles. I think there is a knee in the fatigue curves, but I do not think there is any evidence that beyond that knee the conditions are such that infinite life results.

A. A. MILNE: The pitting experiments of Evans and Tourret are of considerable interest. We ourselves have been carrying out some experiments on the pitting of balls used in ball bearings where the contact conditions are qualitatively very similar. Under the conditions of our experiments pitting is preceded by both surface cracks and by subsurface cracks forming independently. Examination of sections through the contact area, similar to that shown in Fig 8, of numerous balls after different periods of operation leads us to think that the contact stresses and temperatures cause metallurgical changes in the steel, and that the internal cracks first form in this changed region. These experiments were made under higher stress conditions than would be encountered in practice, and it would be interesting to know whether or not the authors had found any evidence of metallurgical changes accompanying the pitting of gears.

T. B. LANE: Although Cameron has said that many of the features of gear lubrication could be explained by the hydrodynamic theory, there are others which cannot be so explained. Applying Cameron's equation VII to the $3\frac{1}{4}$-inch gear rig, it can be shown that the load as actually carried by the rig may be, particularly at high temperatures, some 50 times the load predicted by theory. Some allowance can be made for the effect of pressure on viscosity, which will give an increase of about $2\frac{1}{2}$ times, but this increase will be offset to some extent by the effect of the temperature rise due to heating in the contact. Disregarding this latter effect, however, there is a factor of about 20 still required to reconcile theory and practice. Maybe the effect due to the distortion of the tooth producing a more slender wedge would help, and it would be interesting to know whether Cameron has any idea of the magnitude of the increase to be expected in this way.

Another feature of gear lubrication which seems opposed to the hydro-

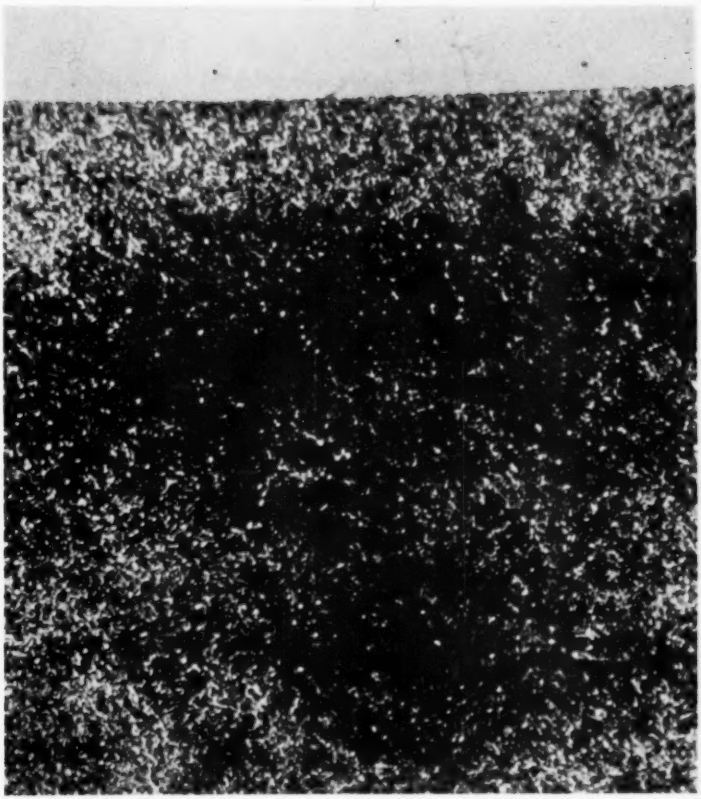

FIG 8

[*To face p.* 686.

dynamic view is the effect of temperature. The viscosity of the oil may be changed by a factor of 10 by the appropriate increase in temperature, and yet the load carried is hardly affected at all. This is difficult to explain, but it appears that the load carried depends more on the constitution of the oil than on its viscosity.

Turning to one alternative to true hydrodynamic lubrication, *i.e.*, boundary lubrication, difficulties are still found, because Bowden * has recently pointed out that light mineral oils at 90° C (the temperature at which many tests are carried out on the rig) have no lubricating ability at all; yet if the teeth of the gears on the test rig are examined just before scuffing the original grinding marks can still be seen. Thus it appears that the lubrication is very effective. It seems to me that there is some other mechanism at work in providing the film. I wonder if the film is supplied by the oil which is taken into the contact by the depressions in the surface and not by that dragged in by shearing forces and if the film is formed by oil expelled from the tiny reservoirs under the action of pressure and heat?

DR CLAYTON: I would not like the point about the possible benefits of vibration of the teeth to be lost sight of. McEwen has already pointed out that the vibration could be obtained other than through torsional oscillation of the system. Good effects result with journal bearings due to build-up of the film, and there are probably many lubricated contacts in practical mechanisms that benefit in the same way.

I support Cameron concerning the value of mathematics, but there are provisos: they must be correct, and the words that go with them must be equally clear and unequivocal.

V. H. BRIX (written communication): I should like some assurance that the viscosity term in the Reynolds hydrodynamical equation can be treated as a variable, for Cameron and McEwen regard viscosity as varying rapidly with pressure. Also, Cameron refers to calculations in which viscosity was taken to vary with the y direction.

The Reynolds equation was originally intended to apply to cases of film lubrication between nearly parallel boundaries, and the paper by Reynolds clearly defined the assumptions and limitations involved. Furthermore, the Reynolds equation is derived from the hydrodynamical equations of viscous flow which were originally deduced by Navier, Poisson, and Stokes. These hydrodynamical equations embody the coefficient of viscosity as a constant term which, surely, implies that its differential coefficient and second differential coefficient with respect to space co-ordinates should be negligible in comparison with the rates of distortion of the viscous fluid. It is, of course, customary to regard the viscosity of lubricant as a variable in the x direction due to heating of the fluid during its progress along a bearing. This is valid because the rate of change of viscosity in this direction is very much less than the distortion du/dy. When, however, we are faced with terms in which viscosity varies exponentially with pressure or, indeed, directly with the space co-ordinates in such a

* Bowden, F. P., "The mechanism of boundary lubrication," *Proc. 3rd Wld Petrol. Congr.*, 1951, **VII**, 328.

manner that its second differential coefficient is finite, the validity may be open to question. Cameron's equation for dp/dx may therefore not be true.

If, however, the validity of these expressions is not open to doubt, I would like to point out a possible explanation of the difference in performance of various lubricants which possess similar bulk viscosity.

We might regard a " good " oil, " good " in the sense of ability to carry high loads, as one which is composed, in effect, of an inactive bulk phase, the solvent, and a relatively active dissolved phase composed of large polar molecules. The latter phase, the solute, will tend to be concentrated at the boundaries of the oil film. The metal surfaces might be expected to carry the highest concentration of the solute, indeed there may be a condensed film of polar molecules, a condition consistent with the highest concentration possible.

According to well-known physical laws, the viscosity of such a solution varies with the concentration of the dispersed phase, and the viscosity at the boundaries would greatly exceed the bulk viscosity of the oil. An adsorbed–condensed-type film adhering to metal surfaces would indeed possess the viscosity of the solid state.

The viscosity of oil trapped between metal surfaces might be regarded, then, as a variable, tending towards infinite value at the surfaces. " Good " lubricants may evince this character to a significant extent, whilst " bad " oils may be " bad " by virtue of not possessing viscosity rise at the boundary surfaces.

One of the difficulties of accepting the viscosity–pressure theory as the key to the gear-lubrication problem, is the fact that castor oil, a " very good " oil, shows less pressure–viscosity effect than mineral oils, many of which are quite " bad " lubricants.

It is suggested that a " surface-viscosity " theory, as outlined above, may prove more adaptable to cases of this kind. With castor oil we have the familiar case of the active fatty acid, ricinoleic acid, doing most of the work. Highly viscous surface layers containing this disperse phase may contribute sufficient measure of viscosity to the film, as a whole, as to influence the load-carrying ability of the oil.

In the paper by Barwell and Milne an experiment is described in which high fluid pressures were shown to exist in the oil film between a disk and a pad under pure sliding conditions. It was mentioned that " a substantial portion of the load may be carried by fluid pressure " implying that surface contacts between the moving metals may have taken place. If, however, the regime was one in which such contacts took place, it would appear that a film of mean thickness of 10^{-5} inch or less is sufficiently thick to fill a pressure gauge in a reasonably short space of time. If it is assumed that the presence of the hole and the gauge had little effect on the performance of the pad, it follows that the mass flow of fluid into the gauge should be small in comparison with the net mass flow under the pad as a whole. If the latter is taken as Q per unit width,

$$Q = \frac{Uh}{2} - \frac{h^3}{12\mu}\frac{dp}{dx} \text{ (notation as usual in bearing literature)}$$

(or in its form applicable to the pad in question), it would be of interest to

compare this, for various values of h, with the flow into the gauge. It is suggested that the answer might show that h, the film thickness between pad and disk, might be of a greater order of magnitude than 10^{-5} inch, most probably 10^{-4} inch.

Additionally, the question of surface contacts might be settled by the provision of some electrical or thermo-electric device sensitive to contacts between the moving metals.

The writer could not obtain satisfactory evidence of contact when studying conditions very similar to this one, and was led to believe, as a result of this as well as other experimental evidence, that the film was too thick to give metallic contacts. Failure, *i.e.*, scuffing, seemed to take place only after the film had collapsed.

L. S. EVANS (written contribution): In their paper Barwell and Milne refer to the change in the surface of steel after scuffing, and suggest that the hard surface constituent has been formed by quenching from a high temperature. It may be that scuffing begins when the temperature is sufficient to initiate the transformation of the steel to the austenitic form; the temperature at which this occurs is greatly influenced by the alloying elements in the steel, and the rate at which it proceeds with increasing temperature increases with carbon content.

Of the alloying elements mentioned in the report summarized by Mansion, chromium, molybdenum, and vanadium raise the critical transformation temperature, whereas it is lowered by nickel and manganese. The magnitude of transformation temperatures can be compared approximately by taking from S curves the temperature at which transformation from austenite begins and in Table C these figures are given for the four-non-case-hardening steels covered in this report.

TABLE C

Steel No.	Alloying Elements, %	Scuffing load		Trans. temp, °C
		Oil 1	Oil 2	
30A/1	Ni 4·35 Cr 1·29 Mo 0·08	3500	2210	670
18/2	Ni 0·24 Cr 0·81	4700	4390	710
24	Ni 1·46 Cr 1·12 Mo 0·23	4800	3430	710
40C	Ni 0·21 Cr 3·49 Mo 0·97 V 0·26	8850	7330	730

Thus for these four steels the higher the transformation temperature the greater the scuffing load.

Furthermore, this criterion would explain why a case-hardened En 18

steel has a lower scuffing load than a through hardened steel of the same composition; the higher carbon content of the case would decrease the AC_3 temperature, so that for a given temperature between AC_1 and AC_3 the proportion of austenite would be much higher in the case of hardened steel, and so scuffing would occur more easily.

Obviously these experiments are too few for reliable conclusion to be drawn, but this suggested explanation of the effect of alloying elements on scuffing load may be worth consideration if more results become available.

Replies to the Discussion

Dr A. Cameron: In reply to Blok's most interesting and lengthy contribution, I can only say that the computation on the problem by Johannes Dörr at Darmstadt which we are supporting from Pametrada, should give the correct answer. It is very good of him to have taken so much trouble in preparing his contribution.

Clayton has suggested that there may be possible benefits due to the vibration of the teeth. I do not think that in practice this is so, for two main reasons. First, disk machines (which have friction characteristics very similar to gears) running with a vibration between the disks do not appear to give a greater load-carrying capacity than machines running smoothly. Secondly, when a gear is running with a torsional vibration present, which will show itself as a vibration normal, or nearly normal, to the tooth surface, the gear surface is usually more or less damaged, sometimes very severely.

Brix has queried the use of variable viscosity in lubrication theory. I think that the answer is that anything can be allowed to vary, provided the start is far enough back in the equations. In the original Navier–Stokes equations everything varies in all directions. The equation $\frac{dp}{dx} = \frac{\partial}{\partial y}\left\{\eta\frac{\partial u}{\partial y}\right\}$ is quite general, and the assumptions made are in effect only that the flow is laminar, and incompressible, and that inertia effects are neglected. It is all dealt with very well by H. M. Martin (*Engineering*, 30.7.51, p. 101).

The double phase lubricant he postulates is attractive, but I cannot see how it can come about. Where do these polar compounds come from? The same criticism can apply to this idea as to Heidebrock's adsorbed layer. It is very hard to reconcile it with known laws of surface chemistry.

Castor oil is indeed a problem. It has a high viscosity but also a low *pressure* freezing point, and its thermal conductivity is higher than mineral oils. Possibly these taken together may explain its good lubricating power.

Dr F. T. Barwell: The suggestion by Evans that scuffing begins when the temperature is sufficient to initiate an austenitic transformation is an interesting one. Though we feel that it may be an over-simplification, for the lubricant almost certainly affects the scuffing temperature, it certainly merits full consideration.

We fully support Blok's warning against the indiscriminate use of Almen's scuffing criteria of PV and PVT; as we pointed out these are intended to be used under restricted conditions only. The purpose of our experiments, which are still continuing, was directed at determining whether there was, in fact, a criterion that could be used under more general conditions. Of possible criteria, the flash temperature hypothesis, originally suggested by Blok, appears the most promising. The criterion considered most applicable to the experiments described was $t\alpha\mu pv^{\frac{1}{2}}$. Blok has himself shown, in the paper to which he refers, that if v is sufficiently high, as it was in our experiments during the greater part of the stroke, that the various cases be considered—square, circular, or band-shaped, stationary, or moving—reduce approximately to this expression. As the nominal area of contact was independent of load, no further correction to p is required.

We have not found, in our experiments, that the flash temperature criterion $t\alpha\mu pv^{\frac{1}{2}}$ is directly applicable. It is possible that the discrepancies are due to over-simplification of the assumptions made by Blok in deriving his expressions, or alternatively to the approximations made in our application of them.

We agree with McEwen that no one with any familiarity with the problem would deny that rolling action affects the load which can be carried between surfaces in relative motion. Whether or not one takes this into account depends on what one is trying to do. In his own paper, for example, McEwen defines v as the relative velocity and does not find it necessary to refer to any rolling action. In common with most workers on the mechanism of scuffing, we have used a sliding motion with the deliberate intention of simplifying the experiment and its interpretation. The type of damage occurring in that region of lubrication where scuffing occurs is unlikely to be affected by the presence of rolling action, although the absolute value of the load required to cause failure may be so affected.

Brix asks what was the magnitude of the film thickness in the experiment on the generation of fluid pressures between parallel sliding surfaces. That is a matter that has been interesting us also, but we have not yet been able to measure it. Contact resistance measurements under very similar, but not identical conditions, with their limitations, showed that if there were no actual metallic contacts the oil film must have been very thin. As there was present a definite, though slight, amount of wear, it is highly probable that some metallic contact took place. We used the phrase " a substantial portion of the load " because a more definite statement on the proportion of the load carried by hydrodynamic pressures would require integration of a pressure plot over the contact area. As the contact area was only 0·037 sq. in. this would be rather difficult. Further, the pressures recorded must not be regarded in anything but a qualitative manner, for with such small film thickness it is probable that the presence of the pressure tapping affects the pressure distribution, and mass flow calculations would be meaningless. Moreover, if pressures may also be generated by micro-irregularities, as has been suggested in some quarters, the pressure hole we used would obviously not detect them. For the present, therefore, we would prefer to regard these pressures as showing, qualitatively, that mixed lubrication conditions persist almost,

if not quite, to the point of scuffing. The precise mode of generation of these pressures still remains as an interesting field of investigation.

H. D. MANSION (written communication): I should like to make a few comments on the rather controversial subject of the effect of surface roughness on the scuffing of gears or disks. Those who claim that hydrodynamic lubrication is maintained between gear teeth appear to be unanimous in recommending that the gear tooth (or disk) surfaces should be as smooth as possible, whereas those who admit that such surfaces operate largely under boundary conditions, as well as some of those who manufacture gears, claim that some surface roughness, suitably orientated, is an advantage. Cameron does not think that surface roughness can be helpful "when there is a continuous oil film supporting the load," but, whilst the latter condition may well apply to constant-speed disks, it has yet to be shown that gear teeth are separated entirely by such a film. McEwen suggests that the change-over limit from scuffing failure of gear teeth under boundary lubrication to pitting failure under fluid film lubrication can be raised by improving the surface finish of the teeth, but this does not appear to be in accordance with practice as is outlined below. Finch, again, suggests that on a rough surface there is nothing but capillary action to replace the oil where this has been rubbed off the points of contact by the sliding action of a mating surface. This might conceivably be the case if the grooves lie in the direction of sliding, but surely if the grooves lie at right angles to the direction of sliding oil will be picked up by each successive ridge as it passes over the grooves?

Barwell and Milne offer a very acceptable explanation of the reasons why some slight roughening of the surfaces may be beneficial, and this explanation would certainly favour the case of the grooves lying transverse to the direction of sliding. Departing from theory, Stone in Part II of the Symposium points out, as an historical fact, that gear grinders using the formed-wheel process achieved greater freedom from scuffing, and hence higher load-carrying capacity, by making their gears with greater surface roughness, the grooves of the latter being, by this process, at right angles to the direction of sliding. This phenomenon is not a recent finding, but dates back at least as far as 1935, although the change to the very rough 46-grit grinding wheel was not generally accepted until the early 1940s, when the manufacture of aero-engine reduction gears rose to mass-production levels.

Some disk tests made at MIRA (to which Stone refers) indicated that case-hardened disks ground axially on a spline grinder, in imitation of ground gears, carried about twice the load sustained by similar disks ground circumferentially by the normal method. (The motion was a combination of sliding and rolling.) It is thought that this effect could hardly be due to "intra-metallic" phenomena alone (*e.g.*, due to the effects of harsh grinding or the like) and that the oil in the grooves must play an important part when hardened materials slide on one another in this manner, particularly during the running-in process. The above remarks do not apply to worm gearing, for which there seems to be ample practical evidence to indicate that the threads of the hardened steel worm require to be as smooth as possible to give best results against the softer bronze worm wheel. The

contact conditions here, however, are much more closely analogous to those of a steel journal rotating in a softer bearing.

PROFESSOR E. MCEWEN : It has been remarked that no judgment can be passed on the magnitude of a sin without some knowledge of the amount of provocation. My calculations on the load-carrying capacity of the oil film between cylindrical surfaces are prompted by the well-known use of a modified index of 0·8 for the relative radius of curvature in calculating the surface or Hertzian load-carrying capacity of gear teeth. This has often been "justified" by the argument that although the Hertzian elastic load-carrying capacity is directly proportional to the relative radius the capacity of the oil film has been alleged to be proportional to the square root of the relative radius so that the 0·8 index is a compromise between these two. The present investigation and its predecessor * prove that there is no hydrodynamic justification for the modified 0·8 index, and in fact it is now agreed that the relative radius of curvature should appear unmodified in the load-carrying capacity formula and that the known relatively lower capacity of large gears must be attributed to their relative higher pitch line velocities so that the effect can be fully covered by reverting to the pitch line velocity as a speed criterion instead of the BS 436 (1940) use of revolutions per minute.†

Clayton rightly objects to the vague description "gear tooth lubrication," and Barwell is also critical on this point. It is, of course, quite clear that calculations such as mine and Cameron's apply to rolling action, and are valid only on the pitch lines of spur, helical, and bevel gears; they are not applicable to gear teeth having an axial sliding component as in hypoid, spiral, and worm gears, neither are they strictly valid at tip and root on spur, helical, and bevel gears. Nevertheless, the use of rolling disks to simulate gear teeth is legitimate within fair limits even at these points, since the ratio of relative sliding velocity $z|(v_1 - v_2)/(v_1 + v_2)|$ never exceeds unity. The use of pure sliding experiments to represent a phenomenon where there is always some rolling and may be no sliding seems to me far more open to question.

The limits of hydrodynamic lubrication fixed by roughness of the surfaces are readily understood; when, however, the film thickness is reduced to the limit at which the peak pressure becomes infinite and is then further reduced, the pressures in the peak zone become negative on the existing theory, which is clearly absurd. The cause is obviously that the approximations made in deriving the Reynolds equation from the Navier–Stokes (see reply to Brix, *infra*) are no longer valid.

I do not believe Gatcombe's‡ vibration effect is of value in gear teeth, since the amplitudes were of the order 10^{-3} inch, which could only be achieved in gear teeth by torsional vibration of the system or by vibration of the rim as a circular beam; the dynamic increment of load on the gear teeth due to the vibration would, in my opinion, more than cancel out the gain in hydrodynamic capacity.

Tuplin discusses the "slide–sweep" ratio. It is clear that a gear

* McEwen, E. *Engineer*, 1948, **186**, 234, 667.
† Merritt, H. E. *Proc.I.Mech.E.*, in the press.
‡ Gatcombe, E. K. *Trans. A.S.M.E.*, 1951, **73**, 1065.

tooth has a hydrodynamic lubrication load capacity only when in contact with a mate, and it is the characteristics of the pair that determine the capacity. The velocity of the contact line over the individual surfaces is v_1, v_2; then the relative or sliding velocity is $|v_1 - v_2|$ and the mean velocity $\frac{1}{2}|v_1 + v_2|$. These quantities enter into the theoretical equations. It is convenient then to define the sliding and rolling velocities as the difference and sum respectively of the individual velocities and the *specific sliding* $= |v_1 - v_2|/|v_1 + v_2|$. This last appears in Cameron's energy dissipation equation (XI). The specific sliding here defined is zero for pure rolling, is infinite for equal and opposite velocities, lies between 0 and 1 for velocities in the same direction, is unity when one velocity is zero, and exceeds unity for velocities in opposite directions. Elsewhere the proposal has been made to use the ratio $|v_1 - v_2|/v_2$, where v_2 is the smaller velocity numerically; this leads to the absurdity that the Timken machine in which v_2 is zero, but v_1 is not, has infinite slide/roll ratio whilst disks with equal and opposite velocities, which is worse, have a ratio of only 2; the present proposals give the Timken machine a slide/roll ratio of unity.

In the case of worm and other hyperboloidal gears the specific sliding should probably be the ratio of vector difference to vector sum of velocities, but it would probably be necessary also to state the angle between their directions.

Blok is, of course, perfectly correct that the theoretical critical loading with infinite local pressure cannot be approached, since the assumptions underlying the derivation of the equation of motion are no longer valid. I am most grateful to him for pointing out the fact that for normal values of n the power law I used leads to substantially the same result as the exponential law preferred by Cameron. The power law $\mu = \mu_0(1 + p/k)^n$ was used by me because it gave three disposable constants μ_0, k, n, to fit the actual pressure–viscosity relation, whereas the exponential $\mu = \mu_0 \exp(\alpha p)$ only gives two, μ_0 and α; it follows from Blok's analysis, however, that the two laws are interchangeable with $\alpha = (n - 1)/k$.

Brix suggests that the modified Reynolds equation (1 of my paper) is not valid for variable μ. It may, however, be shown to be applicable within the limits of our problem as follows:

The equations of motion are *

$$\rho \frac{Du}{Dt} = \rho X + \frac{\partial pxx}{\partial x} + \frac{\partial pxy}{\partial y} + \frac{\partial pzx}{\partial z} \quad . \quad . \quad . \quad . \quad \text{(i)}$$

If μ is a function of pressure only, we may still write

$$\left.\begin{aligned} p_{xx} &= -p - \tfrac{2}{3}\mu\theta + 2\mu\partial u/\partial x;\ \theta = (\partial u/\partial x + \partial v/\partial y + \partial w/\partial z) \\ p_{xy} &= \mu(\partial v/\partial x + \partial u/\partial y) \end{aligned}\right\} . \quad \text{(ii)}$$

and substituting in (i) we have

$$\rho \frac{Du}{Dt} = \rho X - \frac{\partial p}{\partial x} + \tfrac{1}{3}\mu\frac{\partial \theta}{\partial x} + \mu\Delta^2 u - \tfrac{2}{3}\theta\frac{\partial \mu}{\partial x} + 2\frac{\partial u}{\partial x}\frac{\partial \mu}{x} + \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)\frac{\partial \mu}{\partial y} + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial z}\right)\frac{\partial \mu}{\partial z} \quad . \quad . \quad \text{(iii)}$$

* Lamb, H. "Hydrodynamics," C.U.P., Ch. XI.

We proceed to make the usual assumptions of film flow, negligible effects, and maximum Mach number $\ll 1$ so that compressibility can be neglected. The three equations (iii) then become :

$$\frac{\partial p}{\partial x} = \mu\Delta^2 u + 2\frac{\partial u}{\partial x}\frac{\partial \mu}{\partial x} + \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)\frac{\partial \mu}{\partial y} + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)\frac{\partial \mu}{\partial z} \quad . \quad . \quad \text{(iv)}$$

Since the problem is two dimensional (*i.e.*, neglecting side leakage) we may write all differentials with respect to y as zero. Further, if the film is thin, having maximum thickness small compared with the relative radius of curvature, we have

$$\frac{\partial^2 u}{\partial x^2}\Big/\frac{\partial^2 u}{\partial z^2} \sim \frac{w^2}{u^2}, \text{ i.e., } \frac{\partial^2 u}{\partial x^2} \ll \frac{\delta^2 u}{\partial z^2}.$$

Similarly, $\partial^2 w/\partial x^2 \ll \partial^2 w/\partial z^2$; $\delta w/\partial x \ll \partial u/\partial z$. Omitting the small order terms equations (iv) reduce to :

$$\left.\begin{aligned} \frac{\partial p}{\partial x} &= \mu\frac{\partial^2 u}{\partial z^2} + 2\frac{\partial u}{\partial x}\frac{\partial \mu}{\partial x} + \frac{\partial u}{\partial z}\frac{\partial \mu}{\partial z} \\ \frac{\partial p}{\partial y} &= 0 \\ \frac{\partial p}{\partial z} &= \mu\frac{\partial^2 w}{\partial z^2} + 2\frac{\partial w}{\partial z}\cdot\frac{\partial \mu}{\partial z} + \frac{\partial u}{\partial z}\frac{\partial \mu}{\partial x} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad \text{(v)}$$

in which the second and third terms are due to the variable viscosity effect. Now for a thin film of thickness h, $\dfrac{\partial u}{\partial x}\Big/\dfrac{\partial^2 u}{\partial z^2}$ is of order of magnitude $(u/l)/(u/h^2) \sim h^2/l$, where l is a characteristic linear dimension of the system and is large compared with h. Further, *excluding infinities of p and* μ, $(\partial\mu/\partial x)/\mu$ is of order of magnitude $(\mu/l)/\mu$, and hence $\dfrac{\partial u}{\partial x}\dfrac{\partial \mu}{\partial x}\Big/\mu\dfrac{\partial^2 u}{\partial z^2}$ is of order of magnitude h^2/l^2, *i.e.*, $2\dfrac{\partial u}{\partial x}\dfrac{\partial \mu}{\partial x} \ll \mu\dfrac{\partial^2 u}{\partial z^2}$. Similarly, $\dfrac{\partial u}{\partial z}\dfrac{\partial \mu}{\partial z} \ll \mu\dfrac{\partial^2 u}{\partial z^2}$ and $2\dfrac{\partial w}{\partial z}\dfrac{\delta \mu}{\partial z} \ll \mu\dfrac{\partial^2 w}{\partial z^2}$.

Then

$$\frac{\partial p}{\partial x} = \mu\frac{\partial^2 u}{\partial z^2}; \quad \frac{\partial p}{\partial y} = 0; \quad \frac{\partial p}{\partial z} = \mu\frac{\partial^2 w}{\delta z^2} + \frac{\partial u}{\partial z}\cdot\frac{\partial \mu}{\partial x} \quad . \quad . \quad . \quad \text{(vi)}$$

The expression for $\dfrac{\partial p}{\partial x}$ is of standard form except that μ is not constant. The usual procedure from conditions of continuity therefore leads to

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) - 6U\frac{\partial h}{\partial x} = 0 \text{ (cf *)} \quad . \quad . \quad . \quad . \quad . \quad \text{(vii)}$$

I cannot follow Brix about "good" oils. Castor oil is in my experience not a particularly good lubricant for spur, helical, or bevel gearing

* Michell, A. "Lubrication," Blackie, 1950, p. 72.

with hydrodynamic lubrication; when the scuffing limit is approached hydrodynamic lubrication has ceased. Castor oil is, however, an extremely valuable worm gear lubricant where the specific sliding (my sense) exceeds unity; in such cases the rolling velocity $|v_1 + v_2|$ is small and the hydrodynamic load capacity correspondingly low. Boundary lubrication then supervenes at fairly low loads. In fine, the fact that castor oil exhibits a small pressure–viscosity effect suggests that it would not be a good lubricant from the hydrodynamic point of view; the pressure–viscosity effect is certainly not the key to *scuffing* resistance. It is perhaps relevant to note that silicones, which were found by Borsoff *et al* * to have an unexpectedly low load-carrying capacity, show a pressure–viscosity effect almost identical with that for castor oil.†

E. A. SMITH: It would have given me pleasure to have enlarged upon my remarks on colloidal graphite in gear lubrication, and I can well understand Clayton's desire for more information. However, I have interpreted the terms of reference of the symposium closely, and kept to gear and associated tests. Wider information on the lubricating effect of colloidal graphite has been given (*J. Inst. Petrol.*, 1951, **37**, 329; *I.P. Review*, 1948, **2**, 332).

The results for the Thoma machine are quoted, not because this skew-axis arrangement of cylinders simulates closely a gear tooth action but because it provides rolling and sliding components in friction. Personally, I am inclined to doubt whether any laboratory machine can simulate practical gear conditions.

Cameron has presented an invigorating aspect to a subject which has threatened to become smothered by the mass of theoretical data accumulated over the years. I would like to feel that when one considers hydrodynamic conditions of lubrication one takes into account, at least qualitatively, changes which take place in the oil film when one reaches a certain interfacial temperature, which may be in the vicinity of 50° C. Careful observations of friction on sliding surfaces can reveal a steady increase in value with temperature, until a critical value is reached, when friction falls sometimes abruptly. The fall may be due to the formation of oxidation or polymerization products in the oil film between the rubbing metal faces. It does suggest that the physical condition of the oil film changes due to chemical changes arising from heating effects, while there is also the probability of a chemical or physico-chemical reaction between oil film and surface.

Cameron draws attention to the drop in film thickness with increasing temperature. Could the sudden seizure be due, also, to desorption of the oil film. I sometimes feel that the failure to give due regard to adsorption and desorption isotherms of oils with respect to metals leaves room for anomalies in hydrodynamic studies of oil film conditions. Bowden and his associates, have drawn attention to the necessity for chemisorption or chemical combination of a fatty acid to a metal before adequate lubrication is obtained. In this we have the extreme, at the other end of which lies a non-polar liquid, such as paraffin. As a suggestion for those interested

* *Trans. A.S.M.E.*, 1951, **73**, 687.

† Cf Fig III, 11 (p. 63) of Michell, A., "Lubrication," Blackie, 1950, p. 72, with Fig. 5, Bridgman, P., *Proc. Am. Acad. Arts & Sci.*, 1949, **77**, 115.

in research work, I might offer the thought that we do not possess much information about the adsorption and desorption criteria for various liquids on various metals. Are the adsorption and desorption plots characterized by hysteresis loops?

His reference to work at Pametrada on surfaces, in which the finish marks run at right angles to the direction of motion, reminds me that some work on these lines was carried out in Finch's department about ten years ago, when it was indicated that such a finish direction gave lowest rate of wear, while finish marks parallel to the direction of finish gave maximum rate of wear. This work formed part of a study of surface finish in relation to wear, when it was found that a separating lubricant, such as colloidal graphite, tended to offset the effect of finish, pointing to the dominance of the hydrodynamic factors in wear under the particular operating conditions.

Finch is a distinguished authority on graphite lubrication, and I can appreciate his insistence on the particles of graphite being fine and orientated to permit lubrication. Some years ago I carried out some friction measurements on orientated and disorientated graphite films, and found that whereas coarse particles required a little pressure to orientate, particles of colloidal graphite could be orientated by rubbing the surface lightly with silk. It is interesting also to recall the observations of Trillat and Morel (*Rev. Inst. franç. Pétrole*, 1950, **5**, 33–8), that large flakes do not adhere to metal rubbing faces at anything but the lowest speeds, adsorption by means of microscopically fine particles being necessary to maintain a slippery film in place.

L. S. Evans and R. Tourret: Results of the few tests that we have made confirm Tuplin's remarks on the importance of the metallurgical properties of the bronze used, either for worm wheels or for disks.

Barwell and McEwen both query whether there is a "knee" in the fatigue curve, *i.e.*, whether there is a load below which fatigue phenomena (pitting in this instance) will never occur, no matter how many times the stress is applied. As can be seen from Fig 11 in our paper, we have no practical evidence on this point, and we merely pointed out, in passing, the results of other investigators. If one differentiates between "fatigue" and "corrosion fatigue," then it is usual to consider the former as having a definite limiting stress below which fatigue never occurs, whereas there is no such limit for the latter, so that, in any particular application involving fatigue stresses the presence or otherwise of a corrosive agent would seem to be the criterion which determines whether or not there will be an endurance limit.

The conditions for Cameron's experiments on through-hardened steel appear, from his description of the intensive grain flow, to be very much more severe than those used by us. However, his results do not conflict with our theories, as we hold that whether surface or interior cracks, or both, occur depends on the circumstances of the application considered. Further, it is to be expected that the direction of the cracks on gears will be influenced by the macro-structure produced during forging of the gear blank, whereas this type of structure is not produced in cast disks. It is interesting to hear that Milne has found both surface and subsurface cracks in the ball-bearings; we think that one of the most interesting results of our work is

that surface and interior cracks may, in some applications, appear simultaneously, and we welcome a further observation of their joint appearance. Presumably the metallurgical changes on hardened steel balls described by Milne are temper products, and we would not encounter any such effect in non-ferrous material; the only metallurgical change that we have observed on bronze is that of hardening.

Blok suggests that Karas and Paritsky have theoretically demonstrated that the addition of a friction component to the normal load will decrease the depth of maximum shear stress; however, he later casts doubt on the validity of applying such theoretical treatment. We hold no strong views on this subject, and consider that the matter should be submitted to practical trial. For instance, a series of tests with varying degrees of frictional component would afford valuable information on this point.

Finch's cavitation theory is very interesting, but we are not sure whether he is advancing this theory as an alternative to our work-hardening-crack theories for the pitting of gears and disks. If so, we find it difficult to account for our observed three-fold variation in the time-to-pitting caused by change in viscosity or by the addition of a strong E.P. agent, neither of which we would have expected to affect greatly the amount of air or vapour in solution or suspension in the oil. The cavitation theory is, however, a possible alternative reason for the occurrence of surface cracks which eventually lead to pits by hydraulic propagation.

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Oilfield Exploration and Exploitation

Geology

2004. Gulf secrets sought. D. M. Duff. *Oil Gas J.*, 21.4.52, **50** (50), 96.—Two vessels of the Woods Hole Oceanographic Institution plan a survey of the Gulf of Mexico and Caribbean between 15 Apr. and 18 May, to determine the thickness and character of the sedimentary rocks and the nature of the basement. Refraction profiles and cores will be obtained at twenty-two stations. Large water samples will be taken from the bottom of each of the four deep basins crossed for carbon 14 determination, and study of heat flow through the ocean floor is planned. J. C. M. T.

2005. Early diagenesis of California Basin sediments in relation to origin of oil. K. O. Emery and S. C. Rittenberg. *Bull. Amer. Ass. Petrol. Geol.*, 1952, **36** (5), 735–806.—The statement made by geologists that " the present is the key to the past " applies with special force to the study of sediments. An individual study has been made of the twelve basins between the continental slope and the shoreline of Southern California, resulting in an estimate of the efficiency of conversion of planktonic organic matter into oil.

The generally higher pH and lower organic content at depth may lead to solution of silica at depth and deposition of bedded chert nearer the mud surface. Diagenetic changes of significance in the origin of oil are produced by bacteria. These processes are influenced by the character of the overlying basin water, which in turn is related to the depths of the floor and sill of each basin. E. N. T.

2006. Significance of temperature on foraminifera from deep basins off Southern California Coast. R. W. Crouch. *Bull. Amer. Ass. Petrol. Geol.*, 1952, **36** (5), 807–43.—The effect of depth and temp upon the species of living foraminifera have for some time been the subject of intensive examinations.

Tests were made on cores taken from deep basins, and from the open ocean off the Southern Californian coast. Temp, oxygen content, and salinity were measured at depth in most of these basins. The physical and chemical conditions that existed in the Los Angeles Basin during Pliocene time have been partially reconstructed from these data, but until these conditions can be studied in all of their varying aspects, the many problems of foraminiferal ecology remain unsolved.

A number of excellent illustrations of foraminifera specimens are included in this paper. E. N. T.

2007. Important strike. Anon. *Oil Gas J.*, 21.4.52, **50** (50), 92.—Wheeler Ridge field at the southeast end of the San Joaquin Valley, Calif., which has for twenty-nine years produced from shallow Miocene horizons, may have an important deeper extension. A wildcat ½-mile south of the nearest shallow production has flowed 1170 b.d. through a 17/64-inch choke on a 3 hr test from 157 ft of (believed) Eocene sand at 9757 ft. Tubing pressure was 2800 p.s.i., oil gravity 35° API, and gas production estimated at 1500 M.c.f/day. J. C. M. T.

2008. Reported late Tertiary thrusting in Northeastern Nevada. J. C. Hazzard and W. R. Moran. *Bull. Amer. Ass. Petrol. Geol.*, 1952, **36** (5), 844–56.—In 1950 the presence of low-angle thrust faulting involving late Tertiary, probably early Pliocene, lacustrine sediments in two areas in eastern Elko County, Nevada, was reported. This report deals with subsequent studies from the viewpoint of petroleum exploration.

The lithologic and structural features of the breccia in the Spruce Mountain Area and its distribution and regional topographic relations show that it is an old alluvial fan deposit related in origin to the present fan at the mouth of a large canyon draining the western slope of Spruce Mountain. This breccia rests on the eroded surface of gently tilted Humboldt beds, with complete absence of structural features indicative of low-angle thrust faulting. A thrust was reported ten miles south; here the basal maroon conglomeratic silt unit of the Humboldt is in depositional contact with the Palæozoic. Again, there is complete absence of structural features indicative of major low-angle faulting.

In the Thousand Springs Valley area, the Palæozoic is anticlinally folded, and evidence indicates that these rocks and the Humboldt beds were folded at the same time. Directly south of the anticlinal area, the lake beds are faulted against the older rocks. No features indicative of low-angle faulting were found. E. N. T.

2009. Denver–Julesburg Basin. M. Goodin. *World Petrol.*, Mar. 1952, **23** (3), 77.—Twenty-eight new discoveries were made in the Denver–Julesburg Basin of northeastern Colorado and southwestern Nebraska in 1951. The area is the most active in the Rocky Mountain region, and there is intensive exploration and drilling.

The basin, which extends north to south for over 400 miles and east to west for approx 200 miles, is deepest in a trough east of Denver which extends northwards to the Cheyenne area.

The history of oil development in the basin is described. The first discovery, in 1863, found oil in Pierre shale west of Pueblo, and the field is still producing.

There was temporary activity in the 1930s in northeast and east-central Colorado, but only minor discoveries were made. Large-scale exploration was resumed after the discovery in 1948 of the Gurley field in Cheyenne County by I Egging, which found oil in the First Dakota (D), and the discovery of oil in Logan County. All of the more recent fields are producing from the Lower Cretaceous Dakota–Lakota.

A map of the 1951 discoveries in the Denver–Julesburg Basin and a location map of the Central Logan County area in Colorado are included. C. A. F.

2010. Abandons Venezuela test. Anon. *World Petrol.*, Mar. 1952, **23** (3), 90.—Conquistador CQ-I, near Valera, western Venezuela, has been abandoned as dry at 11,386 ft; another test is to be made about 8 miles to the southwest.

In Guarico, Dakao 8-I, west of the Tucupido field, was completed for production at 4605 ft. Julio 1, 2 miles northeast of Santa Maria de Ipire was abandoned at 4506 ft, and Azul 1, 2 miles north of Tucupido, has been tested at approx 10,000 ft. C. A. F.

2011. Chimire field, Anzoategui, Venezuela. E. L. Moore and J. A. Shields. *Bull. Amer. Ass. Petrol. Geol.*, 1952, **36** (5), 857–77.—In January 1948 the Socony-Vacuum Oil Co. of Venezuela discovered the Chimire field in the Greater Oficina area of the state of Anzoategui.

On 1 Jan. 1951, about 104 wells had been drilled, resulting in a production of more than 21,789,000 brl and an average daily production of about 55,265 brl. The area aggregates about 9000 acres, and is about 10 miles in length and 3½ miles wide; concessions within the field are held by the Socony-Vacuum Oil Co. of Venezuela, Creole Petroleum Corpn, and the Mene Grande Oil Co., C.A., the Mene Grande Oil Co. operating Creole concessions as well as its own.

In the eastern part of the field the Mesa is dissected by the headwaters of the Rio Chimire, giving vertical escarpments up to 50 ft high, and a total relief of over 140 ft. The climate in the Chimire area is semi-tropical with two seasons, the dry season from December to the end of April and the rainy season for the balance of the year.

Dual-zone completions with only one sand in each zone are common practice, and all completions have been through gun perforations. Well spacing is approx one well to 94 acres. Outlets for the field are through a 12-inch Socony line and a 16-inch Mene Grande line terminating at Puerto La Cruz on the Caribbean coast. E. N. T.

2012. The general and economic geology of Trinidad, B.W.I. Pt I. H. H. Suter. *Colonial Geol. Min. Resources*, 1951, **2**, 177–217.—Published information and material from unpublished reports and research by oil companies has been brought together to give a comprehensive account of the general geological features of Trinidad as a background to the technical aspects and economics of the oil industry. Pt 1 deals with geographical features, and under general geology discusses pre-Jurassic and Jurassic, Cretaceous, Tertiary, and Quaternary formations. G. S.

Geophysics and Geochemical Prospecting

2013. Control of variables in the observation of seismic echoes. E. J. Handley. *World Petrol.*, Mar. 1952, **23** (3), 66.—In seismic prospecting there is little control over the creation of the energy of the explosion and none over its transmission, but there are

many variables relating to recording technique which can be controlled in order to obtain optimum conditions for the particular wave motion required. These include: the size, type, and depth of explosive, and the material opposite which it is exploded which determines the size of the zone of pulverization; the degree of confinement about the explosive; the number of geophones in multiple set up and their relative spacing and their total spread length; the use of electric mixing; the location of the instrument spread from the shot-hole; the filter setting; and the position of the geophones and their plant. C. A. F.

2014. Bulletin of main (Polish) petroleum institute, 1951, 1 (1), *Supplement to Nafta*, 1951, **7**, 1–5.—Gives summaries of papers by H. Gorka on "Optimum production rate," and by S. L. Sulimirski and J. Strzetelski on "Experimental geochemical surface measurement using the bitumen index"; also details of new types of sampling gas apparatus designed by J. Strzetelski, and other Polish Institute news. M. S.

2015. Conditions of application of seismic reflection method in Carpathian Foreland. A. Kislow. *Nafta*, 1951, **7**, 33–6.—Deals with practical application of seismic reflection method in Miocene series of "torton" shales with substratum of anhydrite. This provides good reflections.

From the above it is deduced that for every strike it is possible to determine optimum conditions for reflection. Due to small angles of reflection, it is advisable to have the recording apparatuses not far from the shot and fairly close to each other. Local undulations of the rock may cause total disappearance of echo. M. S.

2016. Patents. U.S.P. 2,580,636 (1.3.45; 1.1.52). A. Wolf, assr to The Texas Co. A reflection method of seismic exploration.

U.S.P. 2,581,063 (16.3.50; 1.1.52). W. A. Alexander, assr to Standard Oil Development Co. A geophone having a greater sensitivity to vertical seismic impulses than to horizontal seismic impulses.

U.S.P. 2,581,070 (6.2.48; 1.1.52). R. H. Blood, assr to Standard Oil Development Co. Apparatus for testing a borehole to determine the presence of possible oil-bearing formations.

U.S.P. 2,581,091 (21.10.50; 1.1.52). P. H. Foster, assr to Standard Oil Development Co. A portable continuous line seismometer.

U.S.P. 2,581,349 (3.10.45; 8.1.52). W. M. Barret, assr to Engineering Research Corporation. Electrical method of determining the presence and depth of an underground mass.

U.S.P. 2,581,412 (6.7.48; 8.1.52). G. Herzog, assr to The Texas Co. Geophysical exploration by determining the spectrum of natural neutrons emitted from an earth mass.

U.S.P. 2,582,719 (1.2.50; 15.1.52). G. H. Ramsey, assr to Standard Oil Development Co. A device for testing for the presence of fluids in a formation traversed by a borehole.

U.S.P. 2,583,288 (12.11.48; 22.1.52). J. J. Arps. Logging a borehole or well by injecting a neutron-emitting substance capable of contaminating the earth surrounding the hole. R. C. R.

Drilling

2017. The guard electrode logging system. J. E. Owen and W. G. Greer. *J. Petrol. Tech.*, Dec. 1951, **3** (12), *AIMME Tech. Paper No.* 3222, 347–56.—By means of guard electrodes placed above and below the measuring electrode the resistivity of the formations is measured by a thin disk of current at right angles to the axis of the borehole. The disk thickens more quickly radially for short than for long guard electrodes, and this principle seems to offer a means of controlling the depth of investigation. This system of logging is free from "lag," "plateau," "shadow," and other distortions found in conventional logs. The guard system is satisfactory with conductive muds, since the relative contributions of hole size and invaded zone to the apparent resistivity are reduced when the mud and filtrate resistivities are low.

Full resistivity value is given to a formation within inches of the boundary of another formation of large resistivity contrast. Hence, true lithology is recorded when current disks of 12 inches or less are used.

Various examples are included to illustrate the capabilities of the guard electrode system. Comparisons are made with conventional logs.

The mathematical theory of the guard electrode system is given. G. D. H.

2018. The laterolog : a new resistivity logging method with electrodes using an automatic focusing system. H. G. Doll. *Petrol. Tech.*, Nov. 1951, **3** (11), *AIMME Tech. Paper No.* 3198, 305–16.—The laterolog has a system of electrodes which in essence gives a sheet of current normal to the walls of the hole. An automatic control gives constancy to the current intensity. The thickness of this sheet is dependent on the electrode spacing. This device reduces the influence of the mud column on the resistivity values measured, and hence the laterolog is especially useful in wells containing a highly conductive mud. Provided that a bed is thicker than the sheet of current, the resistivities of the adjacent beds have little effect on the recorded values. The sheet of current is usually a few feet to several ft thick. The sequence of beds is sharply differentiated. Apparent resistivity values close to the true resistivity are obtained, provided that the resistivity of the invaded zone is not greater than the resistivity of the uncontaminated part of the bed under survey.

The laterolog is very useful where high salinity muds have to be used.

Field examples and the principle and technique of laterologging are described. G. D. H.

2019. Interpreting the results of drill stem tests. M. F. Manville and O. W. Ward. *Oil Gas J.*, 21.4.52, **50** (50), 130.—The modern pressure recorder provides a continuous pressure/time record during the whole of a drill stem test from the moment the tool leaves the surface until it is retrieved. Several such diagrams illustrating the effects of tight formations, plugging, and good flowing formations are provided and discussed. In addition to yielding information about reservoir fluids, permeabilities, and pressures, drill stem tests are also of use for locating sections responsible for lost circulation. J. C. M. T.

2020. Obstacles in rotary drilling and their removal. W. Kowalczyk. *Nafta*, 1951, **7**, 12–15 ; 40–3. (*Petroleum Convention, Krosno*, 16.12.50).—Obstacles may prevent completion of a well and/or its abandonment. There are two kinds of obstacles, those caused by breakdown of the string and others, *e.g.*, due to damage to walls. The first accounts for some 90% of the total according to Soviet statistics, many of them due to fatigue. Several methods of removing broken string and bits from the bottom of well are dealt with. M. S.

2021. Effect of pump horsepower on rate of penetration. E. C. Hellums. *Oil Gas J.*, 5.3.52, **50** (52), 248.—In the past pump h.p. has been used mainly to increase rate of mud circulation. By using part of the h.p. to increase nozzle fluid velocity rather than quantity drilling rate is increased and bit life extended. The ability of the fluid jet stream to do useful work depends on the product of the volume circulated and its velocity. Limiting factors in increasing this work are the h.p. capacity of the pumps and the max pressure to which the equipment can be safely stressed. The arguments are supported by a number of drilling records from Arkansas, East Texas, Louisiana, Mississippi, and the Gulf Coast. J. C. M. T.

2022. Science and technique in Soviet Union. Anon. *Nafta*, 1951, **7**, 29–30.—Deals with achievements of Soviet petroleum industry and gives a method for calculating optimum time for drilling without pulling out the bit. M. S.

2023. Own costs in drilling for petroleum. Z. Onyszkiewicz and A. Waliduda. *Nafta*, 1951, **7**, 114–19.—Only 10 to 20% of wells drilled produce crude or gas. Drilling of some is not completed for technical reasons. This must be included in expenditure. Each drilling operation can be split according to expenses and time taken in preparation, drilling, cementing, and other work.

Tables given show improvement over last decade. Analysis of logs shows that losses

can be cut by planning and increased capital investments in lorries, diamond bits, new derricks, and instruments which save time. Crews should be familiarized with progress.

When oil has been struck it is still necessary to invest in the well, so it is advisable to establish the probable figure of production to be expected. If this is done further expenditure on poor yield wells might be cut by sealing up the hole. It is also as well to estimate costs of transportation and choose the most economical way at an early moment. M. S.

2024. Light drilling rigs (portable). T. Bielski. *Nafta*, 1951, **7**, 61–3.—Portable light drilling rig of foreign design is described with sketches. M. S.

2025. Science and technique in Soviet Union (Monthly Review). Anon. *Nafta*, 1951, **7**, 91.—Describes new standardized transportable power and drilling unit of 300 h.p., drill table speeds 30 to 204 r.p.m., together with pumps capable of delivering 31·5 litres/sec at 95 atm. All instruments are placed by the side of the head driller.

The unit can be used to depths of 2000 m. A smaller unit for drilling to 1200 m is also briefly described. These units are now being supplied to Polish industry.

A short note shows how new geological survey in U.S.S.R. revealed large quantities of minerals. M. S.

2026. Wiltshire Oil's " Land barge " completely packaged. R. F. Carlson. *Oil Gas J.*, 28.4.52, **50** (51), 82.—A Wiltshire Oil Co rig operating in the Pembrook area of the Spraberry trend is completely self-contained except for drill pipe and racks, a change house, fuel storage, and some extra water storage, and is moved complete from site to site on tracks. Preparation for moving takes only 1 hr, and drilling can be resumed within 1 hr of lowering on a new site. The actual movement may take 6 to 8 hr on this field.

The unit carries a 129-ft derrick of double bulge design, which permits standing of 20,000 ft of pipe at one time. A complete mud system, except for dry mud storage, is accommodated below the derrick floor in a substructure which holds mud pits of 200 brl and water storage of 120 brl capacity. All mud manifold piping is welded.

The total weight of the unit is about 500 tons. It has so far drilled five 5000-ft, three 7000-ft holes, and one 8500-ft hole. J. C. M. T.

2027. Patents. U.S.P. 2,580,510 (13.5.48; 1.1.52). E. F. Brady, assr to Eastman Oil Well Survey Co. A core-receiving assembly for a core-taking apparatus.

U.S.P. 2,580,544 (30.12.48; 1.1.52). G. Herzog, assr to The Texas Company. An electronic method of locating the positions of couplings connecting adjacent sections of borehole casings.

U.S.P. 2,580,999 (7.3.49; 1.1.52). J. C. Conrad. Drilling mud pressure gauge.

U.S.P. 2,581,721 (11.9.48; 8.1.52). A. H. Sherwood. A photo-electric warning and brake-actuating device for drill-string lifting mechanism for oil well derricks.

U.S.P. 2,582,682 (26.6.52; 15.1.52). R. R. Crookston, assr to Standard Oil Development Co. A motion take-off device for a drilling rig.

U.S.P. 2,582,700 (6.1.49; 15.1.52). M. R. Jones, assr to Standard Oil Development Co. A hanger device for supporting a string of well pipe.

U.S.P. 2,582,718 (26.12.46; 15.1.52). D. Ragland, assr to Standard Oil Development Co. Drill stem testing tool.

U.S.P. 2,582,724 (30.3.49; 15.1.52). A. L. Stone, assr to Standard Oil Development Co. Pipe spinner for rotating a section of pipe about its longitudinal axis.

U.S.P. 2,582,909 (6.9.47; 15.1.52), L. L. Laurence, assr to Standard Oil Development Co. In the process of cementing areas in a cased oil well in the presence of a bentonite drilling mud, the modification comprising circulating a quantity of inhibited bentonite immediately preceding the cement. R. C. R.

Production

2028. Hydrafrac operations in the Spraberry. W. L. Grossman. *Petrol. Tech.*, Nov. 1951, **3** (11), 9–12.—The Spraberry sandstones in the lower Leonard section of the Permian do not outcrop. The " Upper " Spraberry is generally 200 to 250 ft thick,

with three well-developed sandstones with an average net sand development of 90 ft. The " Middle " Spraberry, 300 to 350 ft thick, is primarily shale with less than 50 ft of sand. The " Lower " Spraberry has four sand members with an average net sand development of 170 ft. Both the " Upper " and " Lower " members are very fine-grained sandstones with thin streaks of shale and limestone. There are vertical fractures which extend from sandstone to shale; very little deposition has been noted in these fractures. Shooting with nitro-glycerine was very successful at Spraberry Deep and good at Tex-Harvey. At Tex-Harvey forty-three wells completed by shooting averaged 236 b.d. and ranged 27 to 522 b.d. for reported potential tests, while 125 wells Hydrafrac treated averaged 315 b.d. and ranged 21 to 803 b.d.

Most of the current development is in the " Upper " Spraberry. Spraberry Deep is anticlinal, while Germania to Benedum may eventually connect as a permeability trap. Generally, the Spraberry sands range 0 to 10 mD permeability; occasionally it is 70 to 80 mD; the average may be 1 mD. Average porosities range 10 to 15%; the G.O.R. is 350 to 900 p.s.i. and bottom hole pressure 2100 to 2300 p.s.i. Poor production in drillstem tests is not sufficient to condemn a section.

All Hydrafrac treatments of the Spraberry to June 1951 were done with a packer in the well. These have been of five types, some in open hole and some through perforations. The bulk have been in open hole. The surface pressures applied have been less than is usual for depths of 6500 to 7500 ft, possibly because of the existing Spraberry fracture system. Usually oil is pumped ahead of the Hydrafrac treatment at 600 to 1800 p.s.i. Max treating pressures have been 1500 to 3500 p.s.i., with injection rates of 3 to 4 brl/min. Originally 1500 gal of gel fluid were used; later larger volumes were used. In some cases 750 gal of gel fluid mixed with 400 lb of sand, followed by 1500 gal of crude oil with breaker chemicals, is considered a single batch. Changes have involved larger amounts of sand, and smaller quantities of breaker chemicals.

G. D. H.

2029. Phase equilibria in hydrocarbon–water systems. IV. Vapour–liquid equilibrium constants in the methane–water and ethane–water systems. O. L. Culberson and J. J. McKetta. *Petrol. Tech.*, Nov. 1951, **3** (11), *AIMME Tech. Paper No.* 3201, 297–300.—The equilibrium constants for methane and water and for ethane and water have been calculated from experimental data. The constants are for the temp range 100° to 340° F, and the pressure range 200 to 10,000 p.s.i.a., and these are presented in tabular and graphical form. The constants for ethane are greater than those for methane, while those for the latter are about the same as for a natural gas–water system. Equilibrium constants for water in all three systems are nearly the same over the greater part of the ranges of pressure and temp studied. G. D. H.

2030. An experimental and theoretical investigation of gravity drainage performance. P. L. Terwilliger, L. E. Wilsey, H. N. Hall, P. M. Bridges, and R. A. Morse. *Petrol. Tech.*, Nov. 1951, **3** (11), *AIMME Tech. Paper No.* 3199, 285–96.—The behaviour has been studied of a 14-ft sand column under gravity drainage, and with fluid extraction at controlled constant rates, the fluid saturation being determined by resistivity measurements. The fluid was 0·25N-sodium chloride. The fluid distribution was measured at different times. The recovery to gas break-through was inversely proportional to the rate of drainage. The gravity drainage reference rate had no particular significance from a recovery standpoint. In order to use this rate as a basis for comparing recoveries the relative permeability and capillary pressure characteristics and displacing fluid viscosities must be identical for the systems compared.

The theoretical background of the studies is given, and close agreement was found between experimental and calculated drainage performance, indicating that steady state relative permeability and static capillary pressure data can be used to describe fluid displacement behaviour. There was a wide variation in liquid recovery before gas break-through, with variation in rate of liquid extraction. Calculations show that little additional recovery can be expected from high-pressure gravity drainage between the time of gas break-through and the attainment of such high gas/liquid ratios as to make further pressure maintenance impracticable. G. D. H.

2031. Porosity measurement comparisons by five laboratories. B. J. Datson, R. L. Slobod, P. N. McCreary, and J. W. Spurlock. *Petrol. Tech.*, Dec. 1951, **3** (12), *AIMME Tech. Paper No.* 3231, 341–6.—Ten natural and synthetic core samples of

varying degrees of consolidation have been used to check the reproducibility of porosity measurements by tests in five laboratories. The sizes were $\frac{3}{4}$ to $\frac{7}{8}$ inches long and $\frac{3}{4}$ to $\frac{7}{8}$ inches dia. Measurements were made alternately at a central laboratory and at one of the other laboratories. Gas permeabilities were also made periodically at the central laboratory. The Kobe porosimeter, Boyle's law porosimeters, water and organic liquid saturation methods were used.

The average deviation of porosity values from the mean was $\pm 0{\cdot}5$ porosity per cent. A high permeability, friable specimen had an average deviation of $\pm 1{\cdot}0$ porosity per cent, the greatest difference being 5·6 porosity per cent.

The differences are greater than the individual reproducibilities of the various methods. The bulk volumes were the most reproducible quantities, excluding caliper measurements, which are inherently too large. Generally gas methods gave slightly higher values than liquid methods.

The samples were believed not to react with the fluids used. G. D. H.

2032. Rational exploitation of producing wells. H. Gorka. *Nafta*, 1951, **7**, 43–8.—The liquid level inside the well and the velocity of fluid are the basic data for rational exploitation of a deposit. Systematic determinations should be made to enable changes to be introduced when necessary. The liquid level in the borehole whilst the oil is pumped affects the efficiency of the pump. If this level is higher than one-tenth of the distance from the well head to the pump above the inlet valve the efficiency of the pump could be increased. Methods of determination of this level are given and include: (1) change of apparent wt of a load according to the density of surrounding fluid; (2) balancing of oil column by a column of water through a valve in pump; (3) electrical conductivity; (4) estimation of gas volume and pressure in the well; (5) differential "debitometer"; (6) echometer and its development; (7) the longitudinal wave recorder. In case of the last two it is necessary to know the wave velocity in the gas present in the well. References. M. S.

2033. Possibility of utilization of natural gas reservoir pressure in simple turbines. H. Leskiewicz. *Nafta*, 1951, **7**, 65–9; 102–7.—In a short paper illustrated by graphs the author considers if a simple expansion turbine could be incorporated in series with long-distance mains to utilize the difference between reservoir pressure (40 to 100 atm) and pressure required for transmission (20 atm). Only those wells providing high pressure gas need be connected. Since natural gas forms hydrates at 15° C which are crystalline it must be heated to allow for adiabatic expansion in turbine. Calculations for 1000 litres/hr throughput show how much the gas needs to be heated prior to expansion, assuming $\eta = 0{\cdot}4$ and $\eta = 0{\cdot}6$, power variations from 10 to 80 kW and p_o/p_i from 0·1 to 0·05.

Three methods of gas preheating are considered in detail: (1) a part of the gas is burnt with excess air (complete combustion) to heat the bulk; (2) a part of gas is burnt with 120% theoretical air (incomplete combustion) and 80% of the flue gases is recirculated through the burner; (3) use is made of flue gases from other sources in a preheater before actual combustion of some of the gas to heat the bulk. Preheater temp should be approx 120° C, inlet gas to turbine should be at 450° C for $p_o/p_i - 0{\cdot}05$ (twentyfold expansion in turbine) and exit temp less than 100° C. Heat expenditure for the three systems is graphed and (3) is found most economical, but (2) is very nearly the same. If volume of gas burnt per kWh produced by turbine varies, so will exit temp. These are plotted against inlet temperature and p_o/p_i values at conditions (1) and (2) for $\eta = 0{\cdot}4$ and $\eta = 0{\cdot}6$. The whole process is thought to be highly economical. M. S.

2034. Measuring the water vapour content of gases dehydrated by triethylene glycol. J. Politziner, F. M. Townsend, and L. S. Reid. *Petrol. Tech.*, Nov. 1951, **3** (11), *AIMME Tech. Paper No.* 3200, 301–4.—Usually the water vapour content of natural gas is determined by measuring the dew point of the gas at the transmission system pressure, and then making use of known correlations. The U.S.B.M. Dew Point Tester is commonly used, but when there has been dehydration of the gas by concentrated diethylene glycol–water solutions considerable difficulty is experienced in observing water dew points because of condensation of liquid hydrocarbon and glycol films on the mirror surface of the tester. As a consequence, the actual water content

is lower than indicated by the standard correlation. These difficulties appear not to occur when gas is dehydrated by concentrated triethylene glycol–water solutions. To check this point tests have been made on the U.S.B.M. Dew Point Tester and gravimetrically, the experimental equipment and technique being described in detail. The observations of water content of glycol-dehydrated natural gas were made at pressures ranging 500 to 2500 p.s.i.a., and at temp of 80° and 90°; 95 and 99·5% (by wt) triethylene glycol solutions were used. Comparison of directly observed dew points with those obtained from the gravimetric data by McCarthy's correlation shows differences of generally about 2° F. Lower dew points were obtained with the 99·5% glycol solution.

Three separate and distinct dew points were observed. These differ in appearance. First there is the hydrocarbon dew point; it appears almost immediately after cooling starts. The second appears to be a thin glycol film, accumulating slowly and forming a haze (it has not been observed with commercial dehydration units). The water dew point starts as a dark spot in the centre of the mirror.

Tests indicated that the variable nitrogen content of the natural gas had little effect on its water vapour content. G. D. H.

2035. The calculation of pressure drop in the flow of natural gas through pipe. F. H. Poettmann. *Petrol. Tech.*, Nov. 1951, **3** (11), *AIMME Tech. Paper No.* 3217, 317–26.—An equation has been derived for use in calculating the sand face pressure of flowing gas wells, allowing for the compressibility factor. The variation due to compressibility has been expressed in graphical and tabular form. The equation has been applied to twenty dry gas wells and eleven distillate wells to compute the sand face pressure, and the results agreed closely with observed values.

Further equations are given for calculating static bottom-hole pressures in gas wells, for horizontal gas flow, and the isothermal horsepower needed to compress natural gas.

Each equation is illustrated by a numerical example. Pseudo-critical properties of condensate well fluids and some natural gases are given graphically, and there are also visc data for natural gases. G. D. H.

2036. The effect of withdrawal rate on the uniformity of edgewater intrusion. M. Muskat. *Petrol. Tech.*, Dec. 1951, **3** (12), *AIMME Tech. Paper No.* 3230, 327–30.—Equations are developed for the differential sensitivity of the up-dip invasion of oil horizons of differing permeability to the driving pressure differential. The water–oil interfaces are assumed to advance with sharp fronts, or that the microscopic displacement efficiency is independent of rate and capillary pressure effects. Under such conditions it is inferred that limitation of producing rates will not greatly inhibit the "fingering" and by-passing tendency in the high permeability bands, unless the differential fluid head across the original oil column is in excess of about 80% of the driving pressure differential. The restrictions are practicable only for oil reservoirs and aquifers of high permeability and dip, but the restrictive conditions may develop automatically under combined gas injection and water drive. G. D. H.

2037. Results of gas injection in the Cedar Lake field. R. M. Leibrook, R. G. Hiltz, and J. E. Huzarevich. *Petrol. Tech.*, Dec. 1951, **3** (12), 355–66.—Cedar Lake is a northwest–southeast anticline in the northern part of the Midland Basin, with about 100 ft of closure. Production is from a dolomite zone within the San Andres limestone at depths of 4600 to 4700 ft. Permeability development is erratic, and it was concluded that this might favour fairly even distribution of injected gas throughout the producing zone. The crude has a bubble point of 326 p.s.i.a., but the original reservoir pressure was 1950 p.s.i.a. The dissolved gas is 156 cu.ft/brl., and the oil visc is 2·44 cp at 326 p.s.i.a. Most of the injected gas is from the Homann field, delivered at 325 p.s.i., and injected at up to 1600 p.s.i. Some gas from the Yates sand is injected without compression.

The field was opened in August 1939. Development was slow to 1943, and essentially nil from 1943 to 1948. Subsequently, rapid development virtually doubled the known productive area. The south is reasonably well defined.

In the injection area before the institution of secondary recovery the performance indicated that primary methods would probably give 10 to 12% of the oil in place.

Production was by reservoir fluid expansion. Gas injection began in Apr. 1946, and eight injection wells were in use at the end of that year. The reservoir pressure was then 600 p.s.i.a. When injection exceeded withdrawals reservoir pressure rose, and the rate was then adjusted to maintain 600 p.s.i.a. Original productivity indices were 0·05 to 0·2 b.d/p.s.i. drawdown. Now average productivity is 68 b.d/well, against 44 b.d/well for primary operation. Gas break-through was detected in mid-1948. In five years of injection output is about 400,000 brl greater than expected for primary operation.

The volume of oil initially in place has been estimated reasonably accurately, while the relative permeability relationship is believed to be like that of Slaughter. "Apparent" relative permeability–liquid saturation trends have been computed for various assumed conformance factors, and the apparent permeability trend for the whole area agrees reasonably well with a 60% conformance factor.

Ignoring gravity drainage and capillary pressure, the fraction of gas flowing has been computed for various gas saturations. From these the rate of gas advance has been derived, and from the ratio of the observed to the calculated time of break-through a conformance factor of 50 to 60% seems reasonable. It appears that recovery under gas injection will be attained as quickly as for primary operation, because in the latter much would be obtained with reservoir pressures under 326 p.s.i.a., when flow would be slow. G. D. H.

2038. A reservoir analyser study of the Woodbine basin. R. C. Rumble, H. H. Spain, and H. E. Stamm. *Petrol. Tech.*, Dec. 1951, **3** (12), *AIMME Tech. Paper No.* 3219, 331–40.—A study of the performance of the Woodbine sand in the East Texas basin has been made by means of an electric analyser. Allowance was made for variations in the geometry of the sand. The apparent water compressibility was determined from the analyser constants and network condenser and resistor values that gave a basin pressure distribution consistent with the observed pressure history of the East Texas field, and the pressure drawdown in the basin at the beginning of 1947. The analyser study indicated an effective water compressibility of $5{\cdot}3 \times 10^{-6}$ vol/vol/lb/sq. ft. The measured compressibilities of the water and sand total $4{\cdot}7 \times 10^{-6}$ vol/vol/lb/sq. ft., suggesting that these are the main components of the apparent compressibility. In the vicinity of the East Texas field the apparent permeability of the Woodbine is indicated as 1000 to 1500 mD, decreasing to 100 to 700 mD near the Mexia–Talco fault line. Although the pressure data are meagre in critical areas, there appears to be little effective communication across the fault line.

If the present practice of returning 90% or more of the produced water to the Woodbine is continued, an area-weighted pressure of 990 p.s.i. can be maintained, provided that there is also a gradual decrease in the rate of production (from 320,000 to 230,000 b.d. of stock-tank oil). With this practice it is indicated that there will be sufficient water yield from the aquifer to produce the East Texas field to depletion under water drive. G. D. H.

2039. Condensate reservoirs. W. Kulczycki. *Nafta*, 1951, **7**, 98–102; 128–32; 154–9.—A theoretical analysis of retrograde condensation. Great losses have been suffered due to wrong treatment of condensate reservoirs as gas reservoirs. Exploitation requires a fall in pressure, and the pipe acts as a distillation tower where at lower pressures near its top liquid dissolved in gaseous phase is deposited and forms reflux. By increasing reservoir pressure this can be prevented and gaseous phase is enriched in higher mol. wt. hydrocarbons. This can be done by cycling the gases, C_1 to C_3 to raise production of C_4 and C_5. According to different authors, this should be done when condensate is present in volumetric ratio 1 in 9 to 18 thousands. Cycling is unnecessary when conditions in reservoir are well above or below critical condensation temperature.

Sampling apparatus and points of importance in analysis which must be determined prior to cycling are described. The cycling process and changes to be expected in proportions of gases produced are also described. The life of such an oilfield depends on careful positioning of wells and repressurizing holes, and on quantity or cycle gas.

On the surface separators remove C_{4+} fractions using retrograde condensation and Joule–Thompson effect. One must beware, however, of formation of hydrates of hydrocarbons if gas temp falls below $t = 8{\cdot}9l^{0{\cdot}285}$ (or $\log t = \log 8{\cdot}9 + 0{\cdot}285 \log l$).

It is possible to improve on multi-stage condensation by following it with pressure absorption. Graphical and empirical methods give best absorption pressures and proportions of absorbent. Absorbent is slowly lost due to retrograde evaporation. Stripped gas is then pumped back into formation.

Three Russian and one British sources are given as references. M. S.

2040. Successful water flooding. W. W. Wilson. *Oil Gas J.*, 21.4.52, **50** (50), 141.—Water flooding cannot be carried out with equal success in all reservoirs. Solution gas drive reservoirs offer the best and water drive reservoirs the poorest chances of success. Gravity drainage and gas expansion reservoirs are also unsatisfactory because of difficulties in packing off upper barren sections and the danger of vertical fluid movement. Residual oil saturation should be over 35% of pore space, and connate water saturation should not exceed 45%. Reservoirs with a history of substantial water production should be looked on with suspicion.

Some notes on reserve calculation, production rate, and costs are added.

J. C. M. T.

2041. Patents. U.S.P. 2,580,660 (20.4.45; 1.1.52). O. H. Crabtree. Well pumping apparatus including a sand catcher for sand settling out of the pumped liquid.

U.S.P. 2,581,277 (17.12.46; 1.1.52). G. J. Morgenthaler. Oil filter.

U.S.P. 2,581,337 (10.1.48; 8.1.52). J. V. Lapik. Oil filter.

U.S.P. 2,581,540 (2.9.48; 8.1.52). H. T. Kennedy and W. F. Rogers, assrs to Gulf Research & Development Co. Reducing corrosion in wells.

U.S.P. 2,581,979 (18.7.49; 8.1.52). M. B. Standing, H. S. Yaplee, and M. Macauly, assrs to California Research Corpn. An electrical resistance method of locating a point in a well at which liquid is being lost.

U.S.P. 2,581,984 (18.7.47; 8.1.52). R. G. Taylor, assr to The Guiberson Corporation. Well swab cup. R. C. R.

Oilfield Development

2042. Science and technique in Soviet Union (Monthly Review). Anon. *Nafta*, 1951, **7**, 119–20.—Describes conditions in Boryslaw (Western Ukraine) where at one time, due to exploitation crude deposits were believed to be exhausted. Since 1945 petroleum industry in Western Ukraine has been modernized, output her man and number of workers have risen. (From *Friendship*, x, 11.3.51.) Also describes a simple device for emptying crude oil from pipes when the well-bottom pump vents are clogged with sand from formation. Two drawings. M. S.

Transport and Storage

2043. Insulating wash tanks pays off. D. H. Stormont. *Oil Gas J.*, 2.6.52, **51** (4), 88.—Insulation economics and method of installing an insulating sheeting to wet-oil tanks are given. The material used is Celotex plus a $\frac{3}{32}$-inch coating of 250° F melting point mastic, banded with steel to keep the sheets in place. G. A. C.

2044. Trans Mountain pipeline. Anon. *World Petrol.*, 1952, **23** (5), 64–6.—Construction has started on $80 million, 700 mile long, 24-inch Trans Mountain pipeline between Edmonton and Vancouver with capacity of 75,000 b.d., to be raised ultimately to 200,000. Max elevation is 4000 ft. Completion is scheduled for late1953. Present refinery capacity at Vancouver is 29,000 b.d. E. B.

2045. Considerations on the throughput capacities of (gas) pipeline systems. T. Drys. *Nafta*, 1951, **7**, 69–74.—Practical formulæ are analysed for: (1) detection of faults in pipelines; (2) calculation of throughput; (3) replacement of one pipeline with another; (4) calculation of total throughput of pipelines of various dia and lengths. M. S.

2046. Patents. U.S.P. 2,580,547 (27.12.46; 1.1.52). J. D. Hollcroft. A self-cleaning tank for crude oil.

U.S.P. 2,582,819 (20.3.45; 15.1.52). B. Daily, assr to Shell Development Co. Automatic shut-off valve for a system comprising a tank and a pipeline for removing liquid from the tank. R. C. R.

Refinery Operations

Refineries and Auxiliary Refinery Plant

2047. Post-war refinery construction in foreign lands hits fast pace. Anon. *Refiner*, May 1952, **31** (5), 83–8, 96.—Since 1945 refining capacity outside the U.S.A. has increased by 150% from 2,016,113 b.d. to 5,040,000 b.d. Construction for another 1,131,600 b.d. is now under way. Details of the 1945 and 1951–52 capacity and capacity under construction in countries outside the U.S.A. are given. A. R. H.

2048. T.E.L. plant increases Ethyl's capacity by one-third. Anon. *Refiner*, May 1952, **31** (5), 144–5.—With a predicted production of 80 million lb/year the Houston plant increases Ethyl Corpn's capacity to produce anti-knock compounds by approx one-third. A. R. H.

2049. Producing 90-octane gasoline with Houdriflow. F. B. Boynton. *Refiner*, May 1952, **31** (5), 133–5.—Performance data on a 4500 b.d. Houdriflow unit are presented and the operation of the unit discussed. A. R. H.

2050. Shell Cardon refinery to expand to 170,000 barrels daily. V. S. Swaminathan. *Refiner*, Apr. 1952, **31** (4), 182–3.—The capacity of Shell Cardon refinery is to be doubled this year. Additional facilities are to include a new distillation plant, modification to the existing distillation unit, and a new lubricating oil plant with propane deasphalting, furfural extraction, M.E.K. benzole dewaxing, and clay contacting. A. R. H.

2051. Modern and unique designs featured in Snyder gasoline plant. W. F. Chapin. *Refiner*, Apr. 1952, **31** (4), 119–24.—The gasoline plant is described, the main feature being the de-ethanizer which contains fifty trays and operates at 500 p.s.i.g. A flow diagram and the design conditions are given. A. R. H.

2052. Eastern States builds platformer in record time. H. M. Martin. *Refiner*, Apr. 1952, **31** (4), 170–2.—Eastern States has recently completed a reforming unit which was on stream only six months after the ground was broken and seven months from when the venture was first conceived. Practically all the engineering work was done simultaneously with construction, and this is described. A. R. H.

2053. Costs of production in refining processes. A. Major. *Nafta*, 1951, **7**, 22–6.—Describes several attempts on this problem and shows that costs of services cannot be justly divided. An approx scheme is devised. M. S.

2054. Insulation methods for air-lift T.C.C. unit. S. E. Swenson. *Petrol. Engr*, Apr. 1952, **24** (4), C22–6.—The 15,000 b.d. T.C.C. unit at Sone Ferry refinery, New York, has tailored insulation for both heat conservation and personnel protection. Insulating materials in use are 85% magnesia for surfaces up to 600° F and a combination of an inner layer of diatomaceous silica and an outer layer of 85% magnesia surfaces above 600° F. Double layer application is used wherever possible. Insulation is finished with metal jackets and asphaltic plastic weatherproofing. Details of insulation and finishes are given for all parts of the plant. E. K. J.

2055. Hydrous calcium silicate heat insulation. E. C. Shuman. *Petrol. Engr*, Apr. 1952, **24** (4), C55–62.—The new insulating material is a chemically reacted mixture of lime and silica, with small amounts of asbestos fibre for hinging action, and has smooth white appearance. It is insoluble in water and maintains its strength when wet, it is

incombustible and has low hygroscopicity. The density of the material is 11 lb/ft, average bending strength is greater than 50 p.s.i., and compressive strength is 150 p.s.i. Thermal conductivity is quoted at 0·39 for a mean temperature of 100° F and 0·54 at 500° F, and operating temperature range is 0° to 1200° F. E. K. J.

2056. Insulating cement—a versatile heat saver. Anon. *Petrol. Engr*, Apr. 1952, **24** (4), C75–8.—Modern techniques are described for the application of mineral wool insulating cement: (1) as a finishing material; (2) in combination with other insulation; (3) as the sole insulator with wire netting framework. Advantages of the plastic-like material are fast and easy application to give a good finish with lasting and effective insulation. E. K. J.

2057. Properly designed, thermal insulation can earn big investment returns. J. T. Gall. *Refiner*, May 1952, **31** (5), 165–8.—The economics of heat insulation in oil refineries are discussed. A. R. H.

2058. Compressor cylinders for flexible operation. R. S. Ridgway. *Refiner*, Apr. 1952, **31** (4), 166–9; May 1952, **31** (5), 140–3.—In Pt I the selection of compressor cyls for gas plants is discussed with particular emphasis on the limitations of compressor cyl size. Methods are also discussed of varying the capacity of cyls by the use of clearance pockets to meet varying loads.

In Pt II the effect of clearance control on compressor cylinders is shown. Operating curves for cyls which have been drawn up over a period of time are illustrated, and these can be used to determine the clearance required to maintain a constant load at varying intake pressures. Clearance pockets consist of spherical bulbs attached to the cyl by indicator connexion if no other provisions are made. As many as four pockets are attached to each cyl. A. R. H.

2059. Evolution of centrifugal compressors in the petroleum industry. H. M. Shedd. *Oil Gas J.*, 26.5.52, **51** (3), 188.—A review is given of the adaptation of these compressors over the last ten years to fluid catalytic cracking, gasoline plants, refrigeration, transportation, and for compressing gases in the chemical field. G. A. C.

2060. Operation and maintenance of vertical gas engine driven compressors. J. B. Williams and V. E. Ford. *Petrol. Engr*, May 1952, **24** (5), C26–34.—The development is traced of the use of vertical or angle gas engines for gas compression, and the importance of systematic inspection and maintenance is emphasized. Recommended procedures are outlined for checking and recording the operation of gas engine plants at intervals ranging from each shift to the annual overhaul. Sixteen suggestions are listed for reducing repair and operating costs. E. K. J.

2061. Engineering reference section. G. L. Farrar. *Oil Gas J.*, 19.5.52, **51** (2), 153; 26.5.52, **51** (3), 215; 2.6.52, **51** (4), 109.—Pt 7 concerns the pumping of viscous liquids and illustrates the effect of viscosity on reciprocating pump efficiency.

Pt 8 concerns use of centrifugal pumps for handling viscous liquids. Three curves show conversion of water pumping ratings to viscous oils. An example is worked out.

Pt 9 deals with the pumping of viscous liquids by rotary pumps; and graphs show effect of entrained and dissolved gas on pump displacement. G. A. C.

2062. Engineering reference section. J. H. Keyes. *Oil Gas J.*, 9.6.52, **51** (5), 103.—Pt 10 in this series gives a nomograph for determination of approximate useful or effective horsepower required to move fluid under known conditions. G. A. C.

2063. Nomograph for pump horsepower. Anon. *Refiner*, May 1952, **31** (5), 138.—The horsepower can be determined from the capacity, head, and efficiency. 0 to 2000 gal/min 20 to 300 ft head. A. R. H.

2064. Refiner's notebook. S. Chesler and B. W. Jesser. *Oil Gas J.*, 19.5.52, **51** (2), 161; 26.5.52, **51** (3), 223; 2.6.52, **51** (4), 115; 9.6.52, **51** (5), 111.—Pt 123 describes how to calculate costs for a closed relief-valve discharge system.

Pt 124 gives an example for cost comparison when setting relief valve below vessel design pressure. A table shows effect on min thickness vessels.

Pt 125 concerns cost comparison when using back-pressure relief valves, and a table shows effect in a refinery flare system. A flow diagram is also given.

Pt 126 describes how to calculate pressure drop when back-pressure relief valves are used; equations based on adiabatic flow of ideal gases being employed. G. A. C.

2065. Thermal aspects of furnace design. C. Hulse and R. J. Sarjant. *J. Inst. Fuel*, May 1952, **25** (143), 94.—Modern developments in the application of laws of heat transmission to furnaces are discussed. The problem is treated here as the simultaneous solution of the equations operative in any given case. These equations cover the following factors: (1) direct heat transfer from flame and gas to charge; (2) transfer from gases to inner surfaces of furnace; (3) conduction into wall; (4) radiant heat exchange between charge and inner surfaces of furnace. These equations must be considered in conjunction with factors determining the absorption of heat into the charge and the load conditions as to fuel input and the conditions of combustion. The relative magnitudes of the heat transfer due to radiation and convection are assessed to provide a basis for evaluating the relative contribution of these mechanisms in terms of local gas velocities, etc.

The main conclusions drawn are: (1) at moderate beam lengths and normal velocities the accurate determination of convectional heat transfer is unimportant when gas temp is greater than 1600° F; (2) except at lowest temp and beam lengths gas radiation is an important factor in many furnaces employing so-called forced convection heat: (3) it would be advantageous even at relatively high gas temp or beam lengths if gas velocities well in excess of those normally used could be economically attained. The text includes many graphs. Forty references. D. K.

2066. Relationship of furnace volume and heat release. D. E. Gerstenberger. *Petrol. Engr*, Apr. 1952, **24** (4), C51–4.—The principles of efficient pipe still heater design are outlined, and the importance of uniform heat absorption is emphasized in order to obtain max heat transfer and flexibility of operation. The limiting factor of metal tube temp has not changed significantly during ten years, so that improved operation is possible only by greater uniformity of heat transfer through the available surface in the heater and better utilization of the radiation section. A more uniform rate of heat transfer around the circumference of the tubes is another factor capable of improvement. E. K. J.

2067. Heat transfer in spiral coils. M. A. Noble, J. S. Kamlani, and J. J. McKetta, Jr. *Petrol. Engr*, Apr. 1952, **24** (4), C7–20.—Heat transfer coeff and friction factors are correlated for heat transfer in spiral coils under: (1) turbulent flow conditions between (*a*) water and water, and (*b*) steam and water; and (2) viscous flow conditions between oil and water. Mathematical relations are given for each case. Both heat transfer film coeff and pressure drops for spiral tubes are much higher than the corresponding values for straight tubes. E. K. J.

2068. Progress review No. 19: Gas burners. J. F. Waight. *J. Inst. Fuel*, May 1952, **25** (143), 118.—Gas burners can be classified according to the method of aeration employed. Neat-gas or post aerated are terms used to describe those burners in which all air is supplied as secondary air. A number of gas-fires, cookers, and radiator designs employ this type of burner. The burners used generally consist of a suitably shaped manifold carrying an array of Bray jets. The use of the manifold type of neat-gas burner in boilers, air heaters, soft-metal melters, and furnaces has increased. One recent burner development has been based on the design of the small Bray industrial jet. Very high gas rates are possible with this burner, and an exceptionally high degree of stability is obtained. It can be used in induced-draught air heaters and can be placed in an air stream moving at high velocity without loss of stability. It is known as the Hypact burner.

Other burners of the neat-gas type which have been developed are one in which gas is discharged by a row of metering orifices, another with a number of gas discharge orifices arranged round the periphery of a circular air-intake duct, a third consists of a gas orifice which directs a high-pressure jet of gas on to a flat target of either solid or liquid.

Low-pressure bunsen burners are extensively used under conditions of natural

draught in domestic space-heaters, industrial ovens, vats, and furnaces. Work has been carried out on combustion characteristics of town gas. Two theories have been advanced to explain light-back in bunsen burners.

High-pressure burner design is more critical than the design of the low-pressure burner because the upper and lower limits of stability are closer together. The high-pressure burner requires flame ports with a high resistance to back-fire.

Burners in which combustion occurs within refractory tunnels are discussed. Three methods are in use for producing the stoichiometric mixture used by these burners. Recent developments in tunnel-burner design have been stimulated by a demand for burners suitable for rapid local heating. In the U.S. the demand for high-speed heating has been met by burners operating on premixed gas and air. Examples of this type are the radiant-cup burner which is primarily a radiator and the other is the cell burner designed for convection heating. A radiant-tube burner reported by Robiette is also discussed. Twenty-seven references. D. K.

2069. Tough water-control problems solved at Welder plant. M. C. Forbes. *Refiner*, Apr. 1952, **31** (4), 142–5.—Cooling and boiler feed water from wells contains a fairly high proportion of dissolved solids giving rise to scale formation and corrosion problems. The methods by which these have been overcome are described. A. R. H.

2070. Nomograph for cooling pond area. Anon. *Refiner*, Apr. 1952, **31** (4), 164. A. R. H.

2071. Cooling tower maintenance. H. E. Degler. *Petrol. Engr*, Apr. 1952, **24** (4), C44–50.—The detailed inspection and repair of a cooling tower as outlined results in longer life and lower operating costs. Recommended procedures for pre-starting, starting, operation, and maintenance of towers are given. E. K. J.

2072. Salvage cleaning. V. E. Bowes. *Petrol. Engr*, Apr. 1952, **24** (4), C38–42.—Salvaging equipment for re-use on a refinery is possible if thorough cleaning is carried out of the scrap parts to enable inspection and repair of damage. The recommended procedure is oil and grease removal, paint stripping, descaling and derusting, and rust prevention. Special materials are available for each operation. For large parts a steam gun projects a suitable detergent solution to loosen and remove oil, grease, and dirt. E. K. J.

2073. Patents. U.S.P. 2,581,041 (14.11.47; 1.1.52). H. J. Ogorzaly and W. A. Rex, assrs to Standard Oil Development Co. Apparatus for utilizing heat contained in hot fluidizable powders.

U.S.P. 2,581,134 (15.3.47; 1.1.52). W. W. Odell, assr to Standard Oil Development Co. Apparatus for contacting solids and gases.

U.S.P. 2,582,116 (24.8.48; 8.1.52). R. R. Goins, assr to Phillips Petroleum Co. Pebble heater chamber design.

U.S.P. 2,582,688 (14.6.49; 15.1.52). L. J. Ford, assr to Standard Oil Development Co. A tower for contacting gases and finely divided solids.

U.S.P. 2,582,802 (19.10.45; 15.1.52). C. F. Terrell, assr to Pure Oil Co. Apparatus for incorporating a slurry containing suspended solids into a liquid to be treated thereby.

U.S.P. 2,582,826 (25.5.45; 15.1.52). H. C. Glitsch, assr to Glitsch Engineering Co. A refinery tray assembly for refining towers. R. C. R.

Distillation

2074. Extractive distillation of high-purity aromatics. C. L. Dunn and G. E. Liedholm. *Oil Gas J.*, 9.6.52, **51** (5), 68; *Refiner*, May 1952, **31** (5), 104–8.—The application of the Shell Oil Co.'s new process for production of nitration grade aromatics and maintenance of product purity are described. The process consists of: (1) pre-fractionation of petroleum stock to prepare a fraction containing a single aromatic;

(2) distillation of the fraction in presence of a higher-boiling solvent such as phenol, followed by stripping of aromatic from solvent; and (3) a mild treating to pass stringent acid-wash colour test.

Flow sheets and specifications are given. G. A. C.

2075. Liquid recovery from hydrocarbon gases. T. H. Anderson, Jr., and W. E. Colburn. *Refiner*, Apr. 1952, **31** (4), 137–41. (*California Natural Gasoline Assoc., Los Angeles, Oct.* 1951.)—The processes available for the recovery of liquid hydrocarbons from wet gas include adsorption, condensation, gas fractionation, and absorption. The factors affecting the choice of cycle are discussed and an example given, illustrating how these factors influence the selection of the most economical process. A. R. H.

2076. Panoma plant favours intermediate separation over consecutive topping. I. E. Nutter. *Refiner*, May 1952, **31** (5), 146–50. (*Convention Nat. Gas Assoc. of America, Houston, May* 1952.)—The Panoma Corpn's natural gasoline plant at Hooka, Oklahoma, is described. An economic study of two sequences of fractionation showed that the separation of the products into groups followed by binary fractionations required only 77% of the heat required by removing components consecutively in order of their volatility. Flow diagrams are included. A. R. H.

2077. Conditions of heat transfer during condensation. K. Szadkowski. *Nafta*, 1951, **7**, 15–17.—Describes the mechanism of heat transfer and arrangement of tubes. Due to flooding the coeff of heat transfer falls from the top of condenser to bottom. Ginabat arrangement of tubes is here very helpful. Presence of uncondensable gas, *e.g.*, air in water, will lower the coefficient considerably. Graphs are given. M. S.

2078. Patents. U.S.P. 2,580,646 (14.4.48; 1.1.52). D. H. Belden, assr to Universal Oil Products Co. A distillation tower having a heating jacket and removable, spaced liquid distributing baffles, each having a tubular upper portion and a flared lower skirt portion, located within the column.

U.S.P. 2,580,651 (11.5.49; 1.1.52). D. M. Boyd, Jr., assr to Universal Oil Products Co. Obtaining a uniform temp and composition gradient throughout a fractionation column by controlled heat distribution to the column.

U.S.P. 2,581,051 (1.6.49; 1.1.52). C. W. Smith, assr to Standard Oil Development Co. Method of operating liquid reflux systems in fractional distillation.

U.S.P. 2,581,088 (3.11.47; 1.1.52). L. D. Etherington and H. W. Scheeline, assrs to Standard Oil Development Co. Method of supplying the heat requirements in the fractional distillation of a mixture of normally gaseous hydrocarbons.

U.S.P. 2,583,412 (16.9.47; 22.1.52). C. S. Carlson and P. V. Smith, assrs to Standard Oil Development Co. Extractive distillation of a mixture of alcohols containing from 1 to 5 C atoms wherein vapours of the alcohols ascend countercurrently to a reflux of the alcohols dissolved in a high proportion of a phenol to effect vaporization of the more volatile alcohol. R. C. R.

Absorption and Adsorption

2079. Taylor–Mayfair cycling plant features new instrumentation. J. J. Maurer, Jr., and W. T. Henrickson. *Refiner*, Apr. 1952, **31** (4), 146–7.—Instrumentation on the rich oil heater is described and illustrated. Safety devices include automatic shutdown when the stack temp exceeds a set limit or when the fuel gas fails to ignite when starting up. A. R. H.

2080. The Arosorb process in refinery operations. W. H. Davis, J. I. Harper, and E. R. Weatherly. *Refiner*, May 1952, **31** (5), 109–13; *Oil Gas J.*, 19.5.52, **51** (2), 112. *API Divn of Refining, San Francisco, May* 1952.)—The Arosorb process for the production of nitration grade aromatics is described. The process is a commercial application of the silica gel absorption method for separating hydrocarbon liquids. Details of aromatic product and saturate product are given for the following feed stocks:

catalytic reformate, catalytic gasoline (200° to 300° F), naphtha (300° to 400° F), kerosine, and catalytic gas oil.

Investment and operating costs are discussed. A. R. H.

2081. Patents. U.S.P. 2,582,415 (15.10.45; 15.1.52). W. H. Claussen, assr to California Research Corpn. A continuous adsorption process for the separation of kerosine distillates using a powdered adsorbent material.

U.S.P. 2,582,443 (31.5.47; 15.1.52). C. B. Linn, assr to Universal Oil Products Co. Process for removing ethane from a mixture of C_2 hydrocarbons containing acetylene by selective absorption in a liquid paraffinic solvent.

U.S.P. 2,583,239 (24.6.47; 22.1.52). J. W. Teber, assr to Sinclair Refining Co. Separating CO from gaseous mixtures by selective adsorption using a reduced metal supported on a porous carrier which has been pretreated with H_2S, mercaptans, sulphides, sulphur, or thiophenes.

U.S.P. 2,583,352 (19.6.47; 22.1.52). C. H. O. Berg, assr to Union Oil Co. Continuous selective adsorption process for separating the components of gaseous mixture wherein a powdered adsorbent flows downwards through an auxiliary stripping zone, a cooling zone, an adsorption zone, and a heating zone successively. R. C. R.

Solvent Extraction and Dewaxing

2082. Reduction of processing losses by introduction of new dewaxing processes for paraffin-base lubricating oils. W. Zajezierski. *Nafta*, 1951, **7**, 78–81; 112–14; 147–8.—Heavy lub from primary distillation of paraffin-base crudes has visc 2° to 6° E/100° C and pour point *ca* 35° C. Its wax content is approx 25% in microcrystalline form. By redistillation of this fraction a range of products may be obtained at a cost of 3% total loss. Analysis of such fraction showed that it contains ceresine (of branched-chain or naphthenic character) which even as 1% in straight-chain paraffin wax influences crystalline form to give needle-like microcrystals. One sample had formula $C_nH_{2n+1\cdot26}$ and m.p. 4·4° C, another C_nH_{2n} and m.p. 36·5° C. Both were found in petrolatum.

Properties of paraffin and ceresin wax as listed are similar except for crystal form. They are highly soluble in high and slightly soluble in low mol. wt. hydrocarbons: thus separation from oils is possible using low mol. wt. solvents pure or mixed. Various methods of solvent dewaxing are given, with results. M. S.

2083. Patents. U.S.P. 2,580,750 (9.5.49; 1.1.52). H. W. Fleming, assr to Phillips Petroleum Co. Solvent extraction of oxygenated organic compounds from a liquid hydrocarbon phase resulting from the hydrogenation of carbon monoxide in the presence of a fluidized iron catalyst.

U.S.P. 2,581,573 (9.6.48; 8.1.52). J. L. Biles, H. C. Myers, T. A. Petry, and W. H. Cowell, assrs to Socony-Vacuum Oil Co. Separating wax and oily constituents in a waxy mineral oil by treating the oil with a solid adsorbent material in particle form at a temp below that at which the wax congeals and separating the wax from the adsorbent material whilst leaving the liquid constituents within its pores.

U.S.P. 2,582,675 (9.8.46; 15.1.52). W. Bonotto. Apparatus for counter-current solvent extraction of oil from oil-bearing materials.

U.S.P. 2,582,883 (27.8.47; 15.1.52). H. C. Myers, assr to Socony-Vacuum Oil Co. Treating a mixture of paraffinic and non-paraffinic hydrocarbons contaminated with asphaltic materials with a double solvent mixture comprising phenol, cresols, and water for the non-paraffinic hydrocarbons and propane plus 1 to 20% of an organic carbonate for the paraffinic hydrocarbons and separating the resultant raffinate and extract fractions. R. C. R.

Cracking

2084. Thermofor catalytic reforming reaches commercial stage. J. W. Payne, L. P. Evans, E. V. Bergstrom, and V. O. Bowles. *Refiner*, May 1952, **31** (5), 117–23. (*API Divn of Refining, San Francisco, May* 1952.)—Four commercial T.C.R. units with a

total capacity of 50,000 b.d. are planned for immediate installation. A synthetic chromia–alumina bead catalyst is used which circulates through a moving bed system comprising a reactor and a kiln. An extensive study of the effect of process variables in a 50-b.d. pilot plant has been made using several feed stocks boiling in the 200° to 400° F range. Yields of up to 96% of gasoline with O.N. in the region of 100 with 3 ml T.E.L. are obtained. Commercial units are being designed with a single reactor with gas recycle of approx 6 moles per mole of naphtha operating at a pressure of 175 p.s.i.g. and a space velocity of 0·7. Average reactor temp may range up to 1000° F. A flow diagram is presented. A. R. H.

2085. Thermofor catalytic reforming process regenerates catalyst. Anon. *Petrol. Engr*, Apr. 1952, **24** (4), C79.—Development of the T.C.R. process is the regeneration of catalyst, similar to that carried out in the T.C.C. process, which enables operation to be maintained over long periods for the severest conditions and results in continuous higher octane ratings. Process efficiency is made independent of coke deposits, and wide boiling range stocks can be processed advantageously. The catalyst reported is cogelled chromia–alumina catalyst in bead form, and the amount circulated is about one-tenth of that circulated in the air-lift T.C.C. unit. E. K. J.

2086. Full-scale unit for fluid hydroforming. J. E. Seebold, J. W. Bertetti, J. F. Snuggs, and J. A. Bock. *Oil Gas J.*, 19.5.52, **51** (2), 111.—The first commercial unit to use the fluidized catalyst technique is described. This will be in operation this year at the Destrehan refinery of Pam-Am-Southern Corpn. Unit is designed to process Gulf Coast naphtha for production of high octane aviation or motor gasoline blending stocks. Flow sheets and relative costs are given. G. A. C.

2087. Catalytic cracking of high nitrogen charge stock. Pt II. J. W. Schall and J. C. Dart. *Refiner*, Apr. 1952, **31** (4), 173–6.—See Abs. No. 1633. The cracking characteristics of high nitrogen charge stocks are discussed. Temp has no effect upon the yield of gasoline at constant conversion, recycle operation shows very little yield advantage over a once-through process. The bottoms-free, first pass gas oil has cracking characteristics as good as those of the virgin gas oil from the standpoint of gasoline and coke yields. The best product distribution is obtained by cracking the virgin gas oil in a once-through operation and then separately cracking the first pass catalytic gas oil.

A series of charts are presented for the estimation of product distributions and the effect of process variables when cracking high nitrogen content gas oils. These may be adapted for other high nitrogen charge stocks by establishing the appropriate reference lines. A. R. H.

2088. More catalysis is refining's watchword. A. L. Foster. *Petrol. Engr*, Apr. 1952, **24** (4), C3–6.—A summary is presented of developments in catalytic processes for petroleum refining since the second world war. The use of catalysts gives improved products with properties which are being increasingly predetermined. The transition from fixed to moving catalyst beds is outlined, and subsequent improvements in the fluidized technique are described. Details are given of recent developments which have resulted in more economical plant operation and products of higher specification. E. K. J.

2089. In reforming it's the catalyst that counts. M. J. Fowle, R. D. Bent, B. E. Milner, and G. P. Masologites. *Refiner*, Apr. 1952, **31** (4), 156–9. (*Western Petroleum Refiners' Assoc., San Antonio, Apr.* 1952); *Oil Gas J.*, 26.5.52, **51** (3), 181; *Petrol. Engr*, May 1952, **24** (5), C40–4.—The Atlantic catalytic reforming process which is emerging from pilot plant development to commercial operation is described. This is a continuous non-regenerative process utilizing fixed bed multiple reactors. Details of the performance of the pilot plant on a variety of feed stocks are given. A. R. H.

2090. Increase earnings through coking of residual. R. T. Colquette and C. W. Peters. *Refiner*, Apr. 1952, **31** (4), 160–2.—Coking as a method of increasing gasoline and reducing fuel oil production is considered economically. A case history is presented of the adoption of the coking process to meet the demand for light distillates from a limited supply of crude. A. R. H.

2091. Patents. U.S.P. 2,580,478 (28.5.49; 1.1.52). H. M. Stine, assr to Standard Oil Co. Process comprising reforming naphtha and gas oil fractions to give gasoline and a hydrogen-enriched by-product gas which is used for hydrodesulphurizing the heated gas oil feed stock.

U.S.P. 2,581,163 (28.5.48; 1.1.52). R. T. Bell, assr to Pure Oil Co. Promoting the cracking of hydrocarbon oils by the addition of a small amount of an organic peroxide.

U.S.P. 2,581,670 (19.7.47; 8.1.52). L. S. Kassel, assr to Universal Oil Products Co. Regeneration of fluidized catalyst by passing a regenerating gas stream through a series of superimposed zones in which catalyst is maintained in fluidized condition, maintaining a continuous cyclic flow of catalyst upwardly through the zones and withdrawing regenerated catalyst from the upper zone.

U.S.P. 2,582,016 (14.2.46; 8.1.52). H. A. Dutcher, assr to Phillips Petroleum Co. A continuous process for cracking a propane-rich light hydrocarbon stream over pebbles first at about 2000° F to produce an ethylene rich stream and thereafter at 3000° F to crack the ethylene to acetylene, rapidly water quenching the acetylene rich stream to 900° to 1300° F and thereafter further cooling the acetylene-rich stream and utilizing the evolved heat to preheat a pebble stream.

U.S.P. 2,582,246 (7.9.47; 15.1.52). P. W. Garbo, assr to Hydrocarbon Research Inc. In the conversion of hydrocarbons in the presence of a dense phase fluidized bed of catalyst the improvement consisting in using a powdered catalyst admixed with a heated particulate thermophore having a higher specific heat than the catalyst.

U.S.P. 2,582,722 (3.10.47; 15.1.52). R. E. Schexnailder, assr to Standard Oil Development Co. Preparing silica–alumina gel catalysts for the conversion of hydrocarbons from a silica–alumina hydrosol.

U.S.P. 2,582,976 (2.1.48; 22.1.52). E. W. M. Fawcett and J. N. Haresnape, assrs to Anglo-Iranian Oil Co. Ltd. Preparing a catalyst for the dehydrogenation and cracking of hydrocarbons by stabilizing dehydrated montmorillonite with hydrogen sulphide.

U.S.P. 2,580,806 (21.4.50; 1.1.52). L. E. Malina, assr to E. J. Houdry. Forming a catalyst comprising an active oxide of aluminium, magnesium, beryllium, or thorium on an inert support.

U.S.P. 2,582,099 (9.7.49; 8.1.52). D. G. Braithwaite, assr to National Aluminate Corpn. Preparing silica–magnesia catalyst by mixing silica hydrogel microspheres and MgO or $Mg(OH)_2$ in aqueous slurry under specified conditions and thereafter drying in controlled manner.

U.S.P. 2,582,254 (7.5.49; 15.1.52). E. A. Hunter, assr to Standard Oil Development Co. Impregnating an inorganic gel with a metal oxide by contacting the moist inorganic gel with an organic solution of the desired metal alkoxide.

U.S.P. 2,580,429 (23.9.49; 1.1.52). E. J. Houdry and J. W. Harrison. Process of producing a catalyst structure comprising silica-impregnated alumina on a porcelain support.

U.S.P. 2,580,641 (20.12.38; 1.1.52). J. R. Bates and G. R. Bond, assrs to Houdry Process Corpn. A silica–zirconium oxide catalyst for hydrocarbon conversion comprising a zeolitic nucleus containing at least 85% silica prepared from a coagulum formed by interaction of a soluble silicate and a zirconium salt.

U.S.P. 2,582,428 (17.6.49; 15.1.52). V. Haensel, assr to Universal Oil Products Co. Preparing an alumina–platinum–halogen catalyst by adding a basic precipitant to aluminium chloride, contacting the precipitate with a platinum-containing solution, and thereafter calcining it. R. C. R.

Hydrogenation

2092. Destrehan fluid hydroformer scheduled for operation soon. J. E. Seebold, J. W. Bertetti, J. F. Snuggs, and J. A. Bock. *Refiner*, May 1952, **31** (5), 114–16. (*API Divn of Refining, San Francisco, May* 1952.)—The fluid hydroformer at Destrehan will be the first commercial hydroforming unit to use the fluidized catalyst technique. Naphtha and recycle gas will be contacted with a fluid catalyst at 930° F and 250

p.s.i.g. in a reactor 7 ft dia and 70 ft long. The catalyst will be regenerated at 1100° F and 260 p.s.i.g. The catalyst will be molybdena-on-alumina with particle size distribution similar to that of fluid cracking catalysts. At a feed rate of 2000 b.s.d. the product will be either 87·2 vol-% 95 F-1 O.N. or 74 vol-% C_5's, O.N., etc., depending upon the operating conditions. A. R. H.

2093. Nickel catalysts in hydrogenation of fats and oils. A. E. Bailey. *Industr. Engng Chem.*, May 1952, **44** (5), 990–4.—A review covering the history and technology of oil hydrogenation, Ni catalyst manufacture and utilization. Thirty-one references. E. J. C.

2094. Patent. U.S.P. 2,580,528 (14.6.48; 1.1.52). H. K. Dice, S. B. Jeddries, Jr., and W. D. Hull, assrs to Celanese Corpn of America. Hydrogeneration of a mixture comprising essentially acetone, M.E.K., acetaldehyde, alcohols, oxides, acetals, and higher complex aldehydes with hydrogen in the presence of a nickel catalyst. R. C. R.

Polymerization

2095. Patents. U.S.P. 2,580,490 (30.1.48; 1.1.52). D. C. Walsh, assr to Standard Oil Development Co. Starting up a low temp *iso*-olefin polymerization plant wherein a mixture of olefins in an alkyl halide diluent is polymerized in the presence of a Friedel–Crafts catalyst.

U.S.P. 2,581,094 (29.9.48; 1.1.52). A. H. Gleason and S. E. Jaros, assrs to Standard Oil Development Co. Reducing insoluble solid polymer formation in a process for preparing drying oils by polymerization of a conjugated diolefin in the presence of a hydrocarbon-soluble peroxide catalyst.

U.S.P. 2,581,147 (23.3.50; 1.1.52). H. G. Schutze, assr to Standard Oil Development Co. Low-temp olefin polymerization with aluminium chloride containing metallic aluminium.

U.S.P. 2,581,154 (1.7.48; 1.1.52). D. C. Walsh, Jr., and H. C. Schutze, assrs to Standard Oil Development Co. *tert*-Butyl halide promotors in a process for polymerizing olefins in the presence of a Friedel–Crafts catalyst.

U.S.P. 2,581,228 (15.6.45; 1.1.52). G. C. Bailey and J. A. Reid, assrs to Phillips Petroleum Co. Preparing an SiO_2–Al_2O_3–NiO catalyst for use in the polymerization of olefins.

U.S.P. 2,583,420 (11.7.45; 22.1.52). J. D. Garber, D. W. Young, and W. J. Sparks, assrs to Standard Oil Development Co. Copolymerizing butadiene and di-*iso*butylene in liquid propane in the presence of aluminium chloride and controlling the reaction temp by returning the propane and olefinic material volatilized during copolymerization.

U.S.P. 2,583,504 (8.7.46; 22.1.52.) D. W. Young, W. J. Sparks, and J. D. Garber, assrs to Standard Oil Development Co. Copolymerizing butadiene and an *iso*mono-alkene in an inert diluent-refrigerant in the presence of aluminium chloride, wherein the reaction is carried out in two stages.

U.S.P. 2,580,647 (29.5.48; 1.1.52). M. S. Bielawski, assr to Universal Oil Products Co. Preparing a phosphoric acid/carbon catalyst by heating carbonaceous material with phosphoric acid to form a charred composite, mixing the composite with water, shaping the paste, and calcining. R. C. R.

Alkylation

2096. Patents. U.S.P. 2,581,014 (11.12.45; 1.1.52). M. H. Gorin and E. Gorin, assrs to Socony-Vacuum Oil Co. A continuous process for methylating aromatics in the presence of an acid-treated alumina–silica catalyst.

U.S.P. 2,582,047 (19.12.47; 8.1.52). H. H. Lee, assr to Phillips Petroleum Co. A process wherein the concomitant catalytic alkylation of a paraffinic hydrocarbon with an alkene and of an aromatic hydrocarbon with an alkene containing 3 to 16 C atoms is

carried out in separate zones characterized in that the emulsified reaction effluent from both zones are combined and thereafter treated to separate catalyst and hydrocarbon phases, the catalyst being returned to the reaction zones. R. C. R.

Chemical and Physical Refining

2097. New catalytic desulphurization process. Anon. *Industr. Chem.*, 1952, **28**, 216.—Desulphurization process (Autofining) operates at 50 to 200 p.s.i.g. and 700° to 800° F, using long life regenerative cat. S is converted to H_2S; H_2 is produced and recycled. Products, obtained in 100% by vol yield, have good colour and stability. Fractions up to 400° F F.B.P. are almost completely desulphurized, O.N. is raised 2 to 3, and lead response substantially improved. Kerosine, gas oil, and long distillates may be processed, and data are recorded for S.R. gasoline, naphtha, and aromatic extracts from naphtha and kerosine. E. B.

2098. White mineral oil for use in the hot-oil dyeing process. R. K. Rhodes. *Amer. Dyestuff Rept.*, 1951, **40** (16), 489.—An outline is given of the refining necessary to produce a white oil. It is concluded that the most suitable oil for use in the hot-oil dyeing process is one from which all unsaturated and aromatic hydrocarbons and other impurities have been completely removed. Otherwise the dyestuff may be solubilized or uneven dye effects may occur. It is also desirable that the oil should remain water-white and odourless at the high processing temp recommended, especially as the oil is not entirely removed in subsequent scouring.

The U.S. Pharmacopœia acid test is used to indicate the true degree of refining of the oil in terms of freedom from unsaturated and aromatic hydrocarbons. Although the optimum visc for oils for this application is not yet fixed firmly, there has developed a preference for oils of low visc (80/90 sec Saybolt at 100° F), since these drain better, are more readily scoured out, result in lower residual oil content in the fabric, and are cheaper than high visc oils. Flash-pts should be above 350° F, and distillation range is about 550° to above 700° F, so there is no unusual fire risk. Processing temp are from 215° to 225° F, and the oils should possess good heat stability. C. N. T.

2099. Patents. U.S.P. 2,580,827 (27.5.49; 1.1.52). J. W. Payne, assr to Socony-Vacuum Oil Co. Apparatus for regenerating powdered absorbent which has become deactivated by liquid phase contact with an oil in a treating process followed by solvent stripping of oily constituents.

U.S.P. 2,581,061 (26.11.49; 1.1.52). A. C. Abeel and J. H. Hibshman, assrs to Standard Oil Development Co. Desulphurizing hydrocarbons by passage over a ferric oxide/chromic oxide mixture obtained by treating a 2 to 1 mixture of ferric and chromic chlorides with ammonia and thereafter heating the treated mixture.

U.S.P. 2,581,064 (5.6.48; 1.1.52). F. M. Archibald, assr to Standard Oil Development Co. Treating petroleum distillates in a series of consecutive stages with sulphur trioxide.

U.S.P. 2,581,065 (7.7.48; 1.1.52). W. A. Arnold, assr to Standard Oil Development Co. Purifying *iso*butylene by sulphuric acid extraction of a C_4-hydrocarbon stream containing *iso*butylene, normal butylenes, butanes, and butadiene.

U.S.P. 2,581,102 (25.10.48; 1.1.52). L. P. Hodges, assr to Standard Oil Development Co. Method of removing oxygenated organic compounds from a raw feed naphtha synthesized from hydrogen and carbon monoxide over an iron catalyst by treating with a polyethylene glycol containing alkali metal hydroxide.

U.S.P. 2,581,117 (16.12.49; 1.1.52). R. M. Love, assr to Standard Oil Development Co. Treating a cracked hydrocarbon which has been contacted with a hypochlorite solution with aqueous alkali metal sulphide.

U.S.P. 2,581,135 (12.2.48; 1.1.52). W. W. Odell, assr to Standard Oil Development Co. Process for removing oxidizable sulphur compounds from a hydrocarbon gas mixture by reacting the gas with oxygen in the presence of a fluidized catalyst.

U.S.P. 2,581,139 (12.8.48; 1.1.52). J. Prophet, Jr., assr to Standard Oil Development Co. Removing reactive sulphur from naphthenic acids by an aqueous washing process.

U.S.P. 2,581,344 (4.6.48; 8.1.52). J. R. Anderson, assr to Koppers Co. Inc. Purification of distilled coke-oven benzene containing a small amount of non-aromatic hydrocarbons higher boiling than benzene by adding non-aromatic hydrocarbon material boiling between 75° and 85° C, distilling off the azeotrope of benzene and said hydrocarbon material and separating benzene from said azeotrope by distillation with a polar azeotropic agent.

U.S.P. 2,581,506 (10.1.48; 8.1.52). C. N. Zellner, assr to Tide Water Associated Oil Co. Treating a liquid phase oxidized hydrocarbon mixture to prepare and facilitate recovery of desired esterification products.

U.S.P. 2,581,507 (27.7.49; 8.1.52). C. N. Zellner and F. Lister, assrs to Tide Water Associated Oil Co. Treating a liquid phase oxidized hydrocarbon mixture to increase the recoverability of saponifiable constituents thereof in distillate form.

U.S.P. 2,581,560 (8.11.47; 8.1.52). R. V. Shankland, S. E. Shields, and E. W. Thiele, assrs to Standard Oil Co. Refining a synthetic mixture of hydrocarbons and oxygenated compounds by contacting it with spent silica–alumina catalyst containing between 10 and 25% of alumina and having 10 to 40% of the catalytic activity of the fresh catalyst.

U.S.P. 2,583,083 (13.2.46; 22.1.52). D. C. Bond and N. B. Russell, assrs to Pure Oil Co. Converting mercaptans in hydrocarbon oils to disulphides by treatment with oxygen in presence of caustic alkali and 2 : 5-di-*tert*butyl hydroquinone.

U.S.P. 2,583,136 (23.9.44; 22.1.52). D. C. Bond, assr to Pure Oil Co. Regenerating aqueous alkali metal hydroxide solutions containing mercaptides with oxygen in the presence of 9-anthranol as an oxidation promoter. R. C. R.

Special Processes

2100. Low-cost removal of sulphur compounds from motor fuels. O. J. Zandona and I. C. W. Rippie. *Oil Gas J.*, 2.6.52, **51** (4), 80.—A new electrolytic caustic regeneration process is described. Caustics used to remove H_2S from gasoline are disposed of, but those used in mercaptan absorption are continuously regenerated and returned to extracting columns. The electrolyzer consists of sixty electric cells in series, the caustic solution being regenerated by an applied direct current. Nascent oxygen from anodes oxidizes sodium mercaptides to disulphides, and these are carried from cell to regenerated caustic tank.

Operating data and diagram of electrolyzer plate assembly are given. G. A. C.

2101. Processing sulphur bearing gases. C. L. Blohm. *Petrol. Engr*, Apr. 1952, **24** (4), C68–74.—The economics of sulphur recovery from natural and refinery gases are discussed.

Two curves show the approx relation between plant capacity and (1) plant investment costs and (2) plant operating costs for sulphur recovery processes. E. K. J.

2102. The present position of petroleum hydrocarbon processing to chemical products. W. Grimme. *Brennstoff-Chem.*, 1952, **33** (3–4), 37–48. (*German Mineral Oil and Carbon Chem. Soc, Ruhr Group*, 6.12.51.)—A review of the upward trend of the petrochemical industry from its start in the first world war and the contributory factors. American practice, favoured by an abundance of raw materials and capital, is taken as the basis. Economical, practical, and political factors are discussed in assessing the international position. Details are given of producers, products, capital expenditure, production costs, work's capacities, and tonnages produced. Twenty-three literature references. R. T.

2103. Petrochemicals from water gas. P. W. Sherwood. *Petrol. Engr*, Apr. 1952, **24** (4), C28–35; May 1952, **24** (5), C21–6: Pt 1.—Methods available for the production of water gas are steam reforming, partial combustion of natural gas, and gasification of solid fuels. The first two methods can be carried out thermally or catalytically, and the gasification process may be intermittent or continuous. Continuous gasification can be carried out: (1) by external heating; (2) by circulation of hot gases; and (3) by means of oxygen. Details of conditions for reaction and types of plant are given

for all the above methods. The purification methods for water gas as used in the petrochemical industry are also described.

In Pt 2 details are given of the commercial methods of water gas conversion in order to obtain the optimum ratio of carbon monoxide to hydrogen for carrying out a synthesis. The factors affecting the process are outlined for: (1) the synthesis of ammonia using hydrogen produced from water gas; (2) the synthesis of methanol from water gas. Six processes for the synthesis of ammonia are compared with reference to reaction temp., pressure, and percentage conversion, and the main distinguishing features of each process are given. The synthesis of methanol can also be based on the reaction of carbon dioxide and hydrogen with certain advantages. A two-stage, low-pressure synthesis which results in the formation of methyl formate as an intermediate is also reported, but so far it has not operated as a commercial process. E. K. J.

2104. Isomerization and hydrogenation reactions in a rectification column. L. Berg, P. R. Hippeley, and W. S. Reveal. *Chem. Engng Prog.*, June 1951, **47** (6), 283–6.—It is frequently desirable to carry out concurrently fractionation and catalytic reactions. To this end a scheme was devised whereby a portion of the packing of a rectification column is replaced by a catalyst of a size and shape resembling column packing. Reaction is effected in the catalyst section and equilibrium between reactants and products approached. The remainder of the column removes the lower boiling portion of the product, while the rest of the product, now no longer of equilibrium composition, is returned to the catalyst section of the reflux. Reactions studied include isomerization of paraffins and naphthalenes, and hydrogenation of di-*iso*butylene, mesityl oxide, and nitropropane, using a hydrogenation catalyst. Reactions so far carried out have only been those in which the product is lower boiling than any of the reactants.
J. G. H.

2105. Production of high purity aromatics for chemicals. D. Read. *Refiner*, May 1952, **31** (5), 97–103. (*Mtng API Divn of Refining, San Francisco, May* 1952.)—Pilot plant Platforming and Udex extraction data are presented illustrating the production of high purity aromatics by these processes. Platforming data are presented on two cuts from a straight-run gasoline 150° to 270° F and 150° to 400° F, and the economics worked out for processing 10,000 b.d. gasoline. It is shown that nitration grade benzene, toluene, and xylenes can be produced together with a high octane motor gasoline with a pay-out time of two years. A. R. H.

2106. Ethylene by steam pyrolysis of ethane. R. A. Carpenter and F. C. Fowler. *Refiner*, Apr. 1952, **31** (4), 148–9.—Experimental data are presented which show that ethane can be converted into ethylene in the presence of steam with yields exceeding 60% per pass. An optimum yield of 64% was obtained at 1800° F with a contact time of 0·014 sec. The reactor consisted of a non-porous alundum tube 0·189 inch dia, 4 inch long. Superheated steam served as the heat transfer medium. A. R. H.

2107. New gas-synthesis plant. Anon. *Petroleum*, June 1952, **15** (6), 157.—Describes a run, lasting twenty-five days, which was carried out on the new American Bureau of Mines gas synthesis demonstration plant at Louisiana, Missouri. The German Fischer–Tropsch process is said to have been greatly improved. Advantages include the elimination of the shift reaction formerly considered necessary in preparing synthesis gas and the use of a large capacity reactor with a moving bed of catalyst which facilitates the removal of the large amount of heat evolved without destroying the catalyst.
R. E. P.

2108. Powdered coal gasification. J. J. S. Sebastian. *Industr. Engng Chem.*, May 1952, **44** (5), 1175–84.—A pilot plant is described in detail for manufacture of synthesis gas. Results, tabulated and plotted, indicate the effect of various variables on gasification of different American coals. E. J. C.

2109. Acetylene—the newest petrochemical. R. S. Aries and R. M. Cziner. *Refiner*, May 1952, **31** (5), 127.—The production of acetylene from natural gas and its uses for chemical syntheses are discussed. A. R. H.

2110. Integrated phenol recovery for low-temperature carbonization plants. W. Lowenstein-Lom. *Petroleum*, June 1952, **15** (6), 154–6.—It has been found that the recovery

of phenol from the effluents of low-temp carbonization plants is an economic proposition, particularly where low-grade coals with high yields of tar acids are being treated. An integrated phenol recovery system for this purpose is described. Details are given of the quantities of phenol that may be expected to be recovered from a typical plant, together with the min flow of effluents and phenol-containing oils required for its economic operation. R. E. P.

2111. United Carbon Company: new plant in Louisiana. Anon. *India-Rubber J.*, 1951, **121**, 564.—In view of expected big increases in the use of oil-base furnace black, United Carbon Co. Inc. is building new plant worth well over £700,000 in S. Louisiana. Annual production rate will exceed 30 million lb of oil-base furnace black of the high abrasion type. C. N. T.

2112. New Philblack plant at Avonmouth. Anon. *India-Rubber J.*, 1951, **121**, 630.—In this plant, already in operation for several months, there are two main production streams: high abrasion furnace type 30 million lb/annum, and a medium grade, 20 million lb/annum. Approx 70,000 tons/annum of a special petroleum fraction are needed to keep the two units in full production. Details and a flow-sheet are given of the plant layout, the reactors and the product-cooling method being described. The carbon black collection process is outlined. C. N. T.

2113. Patents. U.S.P. 2,580,403 (30.4.47; 1.1.52). W. P. Burton, H. G. McGrath, and L. C. Rubin, assrs to The M.W. Kellogg Co. Separating organic acids and alcohols from mixtures thereof by means of an aqueous alkali and a water-insoluble solvent.

U.S.P. 2,580,622 (18.10.45; 1.1.52). R. R. Vogt and W. H. Hamill, assrs to E.I. du Pont de Nemours & Co. Storing acetylene or increasing the acetylene content of gas mixtures containing acetylene by dissolving the acetylene in a tri-alkyl phosphate, a tri-alkyl phosphite, or a tri-alkyl phosphine oxide.

U.S.P. 2,580,742 (25.4.49; 1.1.52). T. F. Doumani and R. W. Long, assrs to Union Oil Co. Separating saturated aliphatic and *cyclo*aliphatic vicinal dinitro hydrocarbons from other nitrated compounds by selective oxidation of the other compounds by heating the mixture with nitric acid.

U.S.P. 2,580,766–7 (1.10.48; 1.1.52). E. L. Hall, assr to American Gas Association Inc. Manufacturing oil gas by the pyrolysis of petroleum oil.

U.S.P. 2,583,254 (5.12.47; 22.1.52). A. Clark, assr to Phillips Petroleum Co. Hydrocarbon synthesis catalyst comprising promoted iron formed by reducing a powdered fused mixture of Fe_3O_4, K_2O, Al_2O_3, and CaO.

U.S.P. 2,583,452 (27.11.46; 22.1.52). R. N. Watts and W. E. Spicer, assrs to Standard Oil Development Co. Preparing alkali metal salt promoted synthesis catalyst comprising an iron, cobalt, or nickel oxide by spraying the molten metal in an oxidizing atmosphere.

U.S.P. 2,581,068 (12.8.48; 1.1.52). L. K. Beach, assr to Standard Oil Development Co. Catalytic vapour-phase oxidation of a petroleum hydrocarbon fraction containing substantial amounts of hydrocarbons having the indene nucleus to form phthalic anhydride.

U.S.P. 2,581,118 (9.5.47; 1.1.52). T. C. Main, assr to Standard Oil Development Co. Method of controlling the temp in a fluidized catalyst hydrocarbon synthesis process wherein carbon monoxide and hydrogen are forced upwardly through a catalyst-charged reaction zone.

U.S.P. 2,581,155 (4.1.49; 1.1.152). D. C. Walsh, Jr., assr to Standard Oil Development Co. Recovering solid aluminium chloride catalysed rubber copolymer of *iso*butylene and isoprene from a warm solution of the polymers in a hydrocarbon by precipitation in the presence of zinc stearate as anti-tack agent.

U.S.P. 2,581,398 (9.2.50; 8.1.52). J. Feldman and M. Orchin. Separating α-methyl naphthalene from β-methyl naphthalene by adding 2-amino-3-methyl pyridine to the mixture and distilling under vacuum so that the pyridine compound and the β-methyl naphthalene form an azeotrope.

U.S.P. 2,581,406 (4.5.49; 8.1.52). C. Golumbic and M. Orchin, assrs to U.S. Govt. Separation of individual phenols from a mixture of isomeric, homologous, or other closely related phenols by counter-current distribution employing a buffered aqueous solution having a pH of about 11.

U.S.P. 2,581,413 (8.10.47; 8.1.52). J. C. Hillyer and J. T. Edmonds, assrs to Phillips Petroleum Co. Production of drying oils by condensing an unsaturated glyceride oil with an aliphatic conjugated polyolefin.

U.S.P. 2,581,452 (29.3.47; 8.1.52). E. Solomon, assr to The M.W. Kellogg Co. Dehydrating aqueous solutions of organic fatty acids by adding a water-immiscible alkylamine or alkylated aromatic amine of higher boiling point than the acid to the solution and separating the non-aqueous phase from the aqueous phase.

U.S.P. 2,581,493 (1.10.45; 8.1.52). J. P. Lyon and A. A. Harban, assrs to Phillips Petroleum Co. A process for recovering pure tertiary alkyl mercaptans from an impure mixture containing same by treating the mixture with a zinc or cadmium salt of an organic acid to form the metal mercaptide which is then hydrolysed with mineral acid to liberate the pure mercaptan.

U.S.P. 2,581,508 (13.9.49; 8.1.52). C. N. Zellner and F. Lister, assrs to Tide Water Associated Oil Co. Esterification of a liquid phase oxidized hydrocarbon mixture.

U.S.P. 2,581,789 (11.4.49; 8.1.52). S. E. Forman, assr to National Distillers Products Corporation. Separation of acetone from ethanol by adding hexane to the mixture and distilling off a fraction comprising acetone, methanol, and hexane whilst leaving a residual fraction comprising a substantially pure constituent of the initial mixture.

U.S.P. 2,582,148 (15.5.47; 8.1.52). H. M. Nelly, Jr., assr to J. F. Pritchard & Co. Recovering desired hydrocarbon fractions as liquids from the crude well outflow by first cooling at outflow pressure to strip out part of the hydrocarbons and thereafter expanding the material and refrigerating it to strip out a further fraction, the residual gas being pumped back into the formation.

U.S.P. 2,582,434 (29.5.48; 15.1.52). A. E. Hoffman and H. S. Bloch, assrs to Universal Oil Products Co. Producing a hydrocarbon drying oil by catalytic conjunct polymerization of a non-aromatic unsaturated hydrocarbon having at least 3 C atoms per mol.

U.S.P. 2,582,710 (28.9.46; 15.1.52). H. Z. Martin, assr to Standard Oil Development Co. Treating carbonaceous solids at elevated temp to form volatile products by heating a fluidized bed of the powdered material.

U.S.P. 2,582,711, K. J. Nelson, and U.S.P. 2,582,712, F. A. Howard (17.5.47; 15.1.52), both assrs to Standard Oil Development Co. Carbonization of finely divided fluidized solids.

U.S.P. 2,582,713 (28.11.47; 15.1.52). H. Z. Martin and I. Mayer, assrs to Standard Oil Development Co. Regeneration of Fischer–Tropsch catalyst to maintain the required fluidizable particle size distribution and low carbon concentration in a fluidized catalyst bed.

U.S.P. 2,582,867 (27.5.48; 15.1.52). H. D. Hartough and J. J. Dickert, assrs to Socony-Vacuum Oil Co. Preparation of *N*-alkylidene-(alkoxyaryl) alkylamine by reacting on aromatic hydrocarbon having only an alkoxy substituent with an alkylaldehyde and ammonium halide in acetic acid.

U.S.P. 2,582,920 (7.11.47; 15.1.52). R. T. Businger and H. L. Gerhart, assrs to Pittsburgh Plate Glass Co. Process of obtaining pure di*cyclo*pentadiene from a mixture thereof with styrene, coumarone, or indene, by cracking the di*cyclo*pentadiene and thereafter condensing and polymerizing the *cyclo*pentadiene vapours.

U.S.P. 2,582,938 (21.10.50; 15.1.52). du B. Eastman and L. P. Gaucher, assrs to The Texas Co. Producing synthesis gas comprising carbon monoxide and hydrogen.

U.S.P. 2,583,255 (24.5.48; 22.1.52). A. Clark, assr to Phillips Petroleum Co. Controlling the density of the catalyst in the synthesis of hydrocarbons and oxygenated organic compounds from hydrogen and CO.

U.S.P. 2,583,413 (30.9.46; 22.1.52). P. H. Carnell, assr to Phillips Petroleum Co. Conversion of a mixture of alkenes to the corresponding alcohols by selective hydration by treating alkene mixtures in successive steps with HF having a higher catalytic concentration in each successive step and removing the alcohol produced in each step before passing the reaction mixture to the subsequent step.

U.S.P. 2,583,090 (29.12.50; 22.1.52). J. L. Cost, assr to Elliott Co. Separating natural gas mixtures composed substantially of nitrogen and methane and recovering methane under pressure.

U.S.P. 2,583,134 (22.6.48; 22.1.52). H. V. Atwell, assr to The Texas Co. A continuous process for the conversion of hydrocarbons consisting in passing a hydrocarbon feed stock over a heated metal oxide to form carbon monoxide and hydrogen and thereafter passing this gas mixture over a hydrocarbon synthesis catalyst to synthesize the desired hydrocarbon fractions.

U.S.P. 2,583,164 (8.9.49; 22.1.52). C. W. Watson, assr to The Texas Co. In the catalytic conversion of synthesis gas to form hydrocarbons, oxygenated hydrocarbons, etc., the gas is prepared by contacting carbonaceous material with a readily reducible metal oxide and the reduced oxide is used as the catalyst in the subsequent hydrocarbon synthesis zone. R. C. R.

PRODUCTS

Chemistry and Physics

2114. Reactions of free radicals associated with low temperature oxidation of paraffins. E. R. Bell, J. H. Raley, F. F. Rust, F. H. Seubold, and W. E. Vaughan. *Faraday Soc. Discussions*, 1951, **10**, 242–9.—Alkyl radicals are produced in the first step of oxidn, which react quantitatively with O_2 to yield alkyl peroxy radicals. Processes involved in this conversion of these radicals to primary stable products via alkoxy radicals are explained and examples quoted :—

$$\text{(i)}\ \text{ROO.} \xrightarrow{\text{RH}} \text{ROOH} + \text{R.} \longrightarrow \text{RO.} + \text{HO.}$$

$$\text{(ii)}\ \text{ROO.} \xrightarrow{\text{RO}_2} 2\text{RO} + \text{O}_2$$

$$\text{(iii)}\ \text{ROO.} \xrightarrow{\text{R.}} \text{ROOR} \longrightarrow 2\text{RO.}$$

(iii) occurs only at low O_2 concn. Modes of decomposition of alkoxy radicals are discussed, and order of stability of radicals is given as

$$CH_3O > C_2H_5O > n\text{-}C_4H_9O > iso C_3H_7O > iso C_4H_9O = tert\text{-}C_4H_9O$$

E. B.

2115. Kinetics of oxidation of hydrocarbons in the gas phase. Theory of the low temperature mechanism. M. F. R. Mulcahy. *Faraday Soc. Discussions*, 1951, **10**, 259–65.—Oxidn mechanism of hydrocarbons in the gas phase in region 200° to 300° C is basically similar to that for more reactive hydrocarbons in the liq phase at lower temp. On this assumption, that rate of pressure rise is given by

$$\frac{d\Delta p}{dt} = k[P]$$

evidence for which has been found; the equation

$$\frac{p_{max}}{\Delta p_{max}} = \tfrac{1}{2}(K[RH]_0 - C)$$

is derived where Δp_{max} is pressure increment from beginning of reaction to point at which p_{max} occurs and $[RH]_0$ is the initial concn of hydrocarbon; K is a const including the specific rate of chain initiation by peroxide breakdown, c is const allowing for possibility that the peroxide may also decompose in a manner not giving rise to active radicals. E. B.

2116. Influence of substituents on the oxidation of hydrocarbons. C. N. Hinshelwood. *Faraday Soc. Discussions*, 1951, **10**, 266–8.—Influence of substituents on ease of oxidn is discussed in relation to theories of oxidn and electronic interpretations of reactivity. Methyl groups stabilize the mol to oxidn, the major influences being the tendency for methyl groups themselves to remain intact and the stabilizing effect of electron flow (−I effect) from methyl to the seat of reaction. Substitution of groups (+I type) such as OR, Cl, and NH_2 accelerates oxidn, and the moment $\overset{+}{C}\text{–}\overset{-}{O}$ of the carbonyl group indicates that in ketones there is also a flow of electrons into the substituent. E. B.

2117. The exhaust flaming problem and inhibition of the hydrogen–oxgyen reaction by hydrocarbons. R. R. Baldwin. *Fuel*, July 1952, **31** (3), 312–39.—The flaming problem of aero-engine exhaust is described. The results of engine tests and related research work are given and discussed. These show that the blue component of the flames results from the ignition of the carbon monoxide and hydrogen present in the exhaust from rich mixtures, and that the ignition is markedly susceptible to the action of inhibitors, hydrocarbons being particularly effective. While the luminosity of the flame arises from the combustion of the carbon monoxide present, the kinetics of the ignition may be determined mainly by the hydrogen content. Tests made in a Pegasus single-cyl engine, in which the liquid inhibitor was injected directly into the exhaust pipe, show the inhibiting action of the fluid on the blue component of the flame. Studies of the effect of hydrocarbons, on the second limit, the third limit, and the nitrogen peroxide sensitized ignition of hydrogen–air mixtures indicate that the correlation with engine tests is best in the case of nitrogen peroxide sensitized ignition. This supports the suggestion that exhaust gas ignition may be sensitized by the nitrogen peroxide present in the exhaust. Study of the effect of inhibitors on the second limit suggests that the primary inhibition process consists in the removal of a hydrogen atom by the hydrocarbon. W. H. C.

2118. Heats of combustion and formation of liquid ethylenimine. R. A. Nelson and R. S. Jessup. *Bur. Stand. J. Res., Wash.*, Mar. 1952, **48** (3), 206–8.—The heat of combustion of ethylenimine was measured at 28° C in a bomb calorimeter. The experimental data yielded the value $-\Delta H_c^\circ = 1591{\cdot}36 \pm 0{\cdot}57$ kilojoules per mole for the heat of combustion at 25° C, the reactants being liquid ethylenimine and gaseous oxygen and the products being gaseous carbon dioxide, gaseous nitrogen, and liquid water. By combining this value with data on the heats of formation of gaseous carbon dioxide and liquid water, the value $-\Delta H_c^\circ = 91{\cdot}90 \pm 0{\cdot}59$ kilojoules per mole was obtained for the heat of formation of liquid ethylenimine at 25° C. (Authors' Abstract.) W. H. C.

2119. Mass spectra of deuteroacetylenes, monodeuterobenzene, and deuteronaphthalenes. F. L. Mohler, V. H. Dibeler, L. Williamson, and H. Dene. *Bur. Stand. J. Res., Wash.*, Mar. 1952, **48** (3), 188–92.—The mass spectra of C_2H_2, C_2HD, C_2D_2, C_6H_6, C_6H_5D, $C_{10}H_8$, $C_{10}H_7D$, and for a mixture of $C_{10}D_8$ and $C_{10}HD_7$ have been determined with a Consolidated model 21–102 instrument, and are recorded. In acetylene, benzene, and naphthalene it is possible to measure the isotopic purity at ionizing voltages below the appearance potential of any fragment ions and to correct the observed spectra of monodeutero compounds for isotopic impurity. In C_2HD the probability of removing H as compared with the probability of removing D is 1·92. There is no such selectivity with respect to monodeutero benzene and naphthalene, and it is possible to compute the spectra of the monodeutero compounds on the basis of equal *a priori* probability of removing a single H or D atom. By assuming that equal probability holds for doubly charged ions, the complete doubly charged ion spectra can be computed from the observed half-integer peaks. Similarly, the spectra of $C_{10}D_8$ and $C_{10}HD_7$ are computed from a spectrum of a mixture of the two compounds. W. H. C.

2120. Some accurately measured infra-red wavelengths for calibration of grating spectrometers. E. K. Plyler, N. M. Gailar, and T. A. Wiggins. *Bur. Stand. J. Res., Wash.*, Mar. 1952, **48** (3), 221–7.—The wavelengths of sixty absorption lines in the 1·1- to 2·3-μ spectral region have been determined and are recorded. The methods of measurement are described, calibration being made by using the absorption bands of

methane and water vapour. The methods of superimposing the well-known emission lines of several sources on the absorption lines that were measured are discussed.
W. H. C.

2121. Some properties of organic peroxides. A. C. Egerton, W. Emte, and G. J. Minkoff. *Faraday Soc. Discussions*, 1951, **10**, 278–82.—*n*, m.p., and v.p. of simpler alkyl hydro-, dialkyl, and acyl hydro-peroxides are recorded and compared with literature values. Latent heats were calc from v.p. data measured in a static system and heats of combustion determined in a bomb calorimeter. Estimates for b.p. of ROOR′ are high compared with ROR′; peracids have lower b.p. than the corresponding acids.
E. B.

2122. Nickel catalysts in acetylene and carbon monoxide chemistry. C. McKinley. *Industr. Engng Chem.*, May 1952, **44** (5), 995–9.—Summarizes important applications of nickel as a catalyst in both acetylene and carbon monoxide chemistry. Examples of reactions are quoted, and suggested mechanisms put forward. E. J. C.

2123. Nickel catalysts for hydrocarbon–steam reaction. M. R. Arnold, K. Attwood, H. M. Baugh, and H. D. Smyser. *Industr. Engng Chem.*, May 1952, **44** (5), 999–1003.—Semi-micro apparatus and experimental procedure described for investigating properties of catalysts in relation to reactions of hydrocarbons with steam. Results suggest that catalysts with higher nickel contents have higher activities at 1500° C. It was found that rapid inactivation of commercial catalysts occurred at temp above 1900° F. Carbon deposition on the catalyst is stated to increase with mol. wt. of hydrocarbon and as steam-to-hydrocarbon ratio is reduced. E. J. C.

2124. Raney nickel-catalysed hydrogenation of commercial aldol. C. K. Hancock. *Industr. Engng Chem.*, May 1952, **44** (5), 1003–6.—Catalytic hydrogenation of aldol is one step in the process for synthesizing butadiene from natural gas. Experiments reported were carried out at pressures of 55 to 875 p.s.i. over the temp range 70° to 120° C. As temp increased from 70° to 100° C reaction time decreased, and yields of 1:3-butanediol also decreased slightly. Above 500 p.s.i. increase of pressure had little effect on yield of 1:3-butanediol, but reaction time decreased. Addition of water up to 30% to the adol did not greatly affect yields or reaction time.
Catalyst could be re-used after suitable treatment. E. J. C.

2125. Volumetric contraction in blending of hydrocarbons. E. J. Reeves. *Refiner*, Apr. 1952, **31** (4), 154.—A correlation, for the volumetric contraction in blending, is presented for various liquid hydrocarbons, including ethane, propane, butane, heptane, decane, benzene, naphtha, kerosine, and crystal oil. Up to 30% deviation from ideality occurs in some cases. A. R. H.

2126. West Texas natural gasolines—their compositions and properties. V. R. Ratzlow and A. A. Ruoho. *Refiner*, Apr. 1952, **31** (4), 125–8.—The results of a precise determination of the composition and properties of two West Texas de-ethanized natural gasolines are presented. 1000 gal of each was desulphurized and then debutanized in a forty-plate packed column. About 250 gal of each product was fractionated in a 100-plate column into 0·1% fractions which were analysed. About 95% of the gasoline was identified as pure hydrocarbons. Sulphur contents, O.N.s, and refractive indices of fractions were determined. A. R. H.

2127. Correlating heat quantities of lower hydrocarbons. D. F. Othmer and R. D. Beattie. *Refiner*, Apr. 1952, **31** (4), 129–35.—It is shown that straight lines result when the entropies of gases are plotted against the entropies of similar gases at the same temperature. This has been confirmed for steam, diatomic gases, olefinic and diolefinic gases, acetylenes, and cyclic hydrocarbon gases. For a homologous series of gases at the same pressure the lines pass through the origin and the slopes are equal to the ratios of the isobaric specific heats. By a similar plot at constant pressure, enthalpies of lower hydrocarbons and other gases may also be correlated.
Entropies at different pressures may be correlated by plotting against the entropy of the reference gas at 1 atm giving a series of parallel isobars.
The theoretical basis of these correlations is discussed. A. R. H.

2128. Extractive crystallization—a new separation process, Pt 11. K. A. Kobe and W. G. Domask. *Refiner*, May 1952, **31** (5), 151–7. See Abs. 1679. The thermodynamics of formation and dissociation of urea adducts are discussed and also the composition and structure of thiourea adducts. A. R. H.

2129. Studies in Newtonian Flow. IV. Viscosity vs molecular wt in liquids; viscosity vs concentration in polymer solutions. A. K. Doolittle. *J. appl. Phys.*, 1952, **23**, 418.—Concluding article in series on Newtonian Flow. The mol. wt.-free space function applicable to *n*-paraffins $m = 100$ to $m = 240$ at various temp is further generalized to apply to homologous series of all Newtonian liquids at constant temp. The resulting mol. wt. function loses some accuracy, and the slopes and intercepts lose their physical significance. The generalized logarithmic decrement equation reproduces visc data over extended ranges of mol. wt. (at constant temp) more successfully than the previously proposed expressions. The equation is extended to apply to mixtures of liquids and further to polymer solutions. L. G. H.

2130. Application of the measurement of the dielectric constant and loss angle to the study of the structure of vulcanizates loaded with carbon blacks. P. Thirion and R. Chasset. *India-Rubber J.*, 1951, **121**, 551. (*Symp. of the Instn of Rubber Industry, London, Sept.* 28, 1951.)—In the non-destructive test method used, information is obtained on the colloidal structure of rubbers containing carbon black; the variables investigated are discussed. C. N. T.

2131. Design and construction of a high-speed camera and its application to certain combustion problems. A. F. Howland and M. J. G. Wilson. *Fuel*, July 1952, **31** (3), 274–87.—A description is given of the design and construction of a high-speed Schlieren camera, and of the spark light source which enables photographs to be taken at the rate of up to 10,000 frames/sec. The camera can be constructed from relatively inexpensive materials, and accurate machining of the parts is not necessary. The camera has been used in the study of: (1) the oscillation of the inner cone of a roaring bunsen burner flame; and (2) the explosive combustion occurring in tunnel burners. Preliminary results of these studies are given and discussed. W. H. C.

Analysis and Testing

2132. Review of analytical chemistry. Anon. *Anal. Chem.*, 1952, **24**, 2–131.—This review covers all the recent developments in the various fields of analytical chemistry which have taken place during 1951. J. S.

2133. Petroleum. H. Levin. *Anal. Chem.*, 1952, **24**, 266–70.—This review continues from the previous one. See Abs. 1460—1951. The developments which have taken place in the analysis and testing of petroleum and its products for the past year. 129 references. J. S.

2134. Coatings. T. G. Rockhow and R. W. Stafford. *Anal. Chem.*, 1952, **24**, 232–7.—This review covers chemical and physical analytical methods applicable to organic high polymers and associated oils, waxes, pigments, and plasticizers as disclosed in the literature during 1951. J. S.

2135. Methods of analysis of petroleum gases. M. Tokarzewska. *Nafta*, 1951, **7**, 74–8; 107–12.—Description of apparatus and procedures employed for gas analysis, compiled from U.S., German, and Russian literature. M. S.

2136. Modern techniques in petroleum testing. VIII. Inorganic analysis. B. W. Swanson. *IP Review*, 1952, **6**, 73.—The introduction of leaded gasolines, additive oils, and the use of residual oils as fuels for internal combustion turbines have given new importance to inorganic analysis in relationship to petroleum products. There are three kinds of analysis necessary: (1) additive concentrates, and their blends in oils and fuels; (2) detection and estimation of trace elements in crude and blended oils and fuels, including examination of used oils; (3) engine deposits. Each of these groups needs specialized treatment, but standard methods have been introduced to a limited extent for the first two. Chemical methods are described and details given of the

application of specialized treatments like emission spectroscopy, spectrophotometry, electrical measurements, polarograph and electrometric titration. R. H.

2137. Correlation of infra-red spectra. Paraffins, olefins, and aromatics with structural groups. H. L. McMurray and V. Thornton. *Anal. Chem.*, 1952, **24**, 318–34.—The object of this investigation was to improve the reliability and convenience of structural analyses of hydrocarbons by infra-red spectra and to provide factual data which might eventually aid in understanding the vibrational origin of certain bands. Correlation tables and charts are presented for paraffinic, olefinic, and aromatic structures. The paraffin correlations were deduced from a study of paraffin spectra only; the olefins from the spectra of unconjugated alkyl substituted compounds. The aromatic correlations were obtained partly from the spectra of alkyl aromatics and, where data on hydrocarbons were limited, from a comparison of the spectra of alkyl aromatics with published correlation charts.

Because the wavelength ranges for the correlation bands have been deduced from hydrocarbon spectra only, they should improve the reliability of the structural analyses of hydrocarbons. Often the ranges thus determined are considerably narrower than shown in the data published heretofore. Intensity data are given which are sometimes useful in identifying the structures giving rise to bands. In some cases they make possible estimates of the concentrations of groups. Critical evaluation of the correlations makes the limitations involved in using any specific correlation band more evident. J. S.

2138. A focusing cell refractometer. L. E. Ashman, W. S. Schwartz, and H. E. Jones. *Anal. Chem.*, 1952, **24**, 191–3.—The focusing cell refractometer provides a method for continuously recording the refractive index of transparent liquids. The instrument is not satisfactory for use with opaque liquids, but it is relatively insensitive to changes in colour sample and deposition of material on the cell surfaces. This latter property is of particular importance in dealing with plant streams because of the large amount of impurities usually present. The experimental instrument constructed will operate with a series of three properly designed cylindrical lenses over a range of 1·34 to 1·56 with an accuracy of $\pm 0{\cdot}0002$. The refractometer has particular application to the monitoring of hydrocarbon feed streams such as butadiene and styrene and as an index of product quality. J. S.

2139. Determination of total per cent aromatics in heavy petroleum distillates. Modified method using silica gel adsorption. A. T. Watson. *Anal. Chem.*, 1952, **24**, 507–9.—A method has been developed for determining the total aromatic content of heavy petroleum distillates which is relatively rapid and suitable for routine application. The analysis utilizes the strong affinity of silica gel for aromatics, and is considered accurate to within $\pm 1\%$. While the procedure is based on the same principles as those used by Tipkin *et al*, it has been simplified by eliminating the necessity for refractive index measurements during the determination and by reducing the sample size requirements, resulting in a significant reduction in time requirement, and making it possible for one operator to carry out several analyses simultaneously. The method is particularly useful for studying the effects of process variables in such operations as solvent extractions of gas oils and lubricating oils. J. S.

2140. Identification of alcohols in dilute aqueous solution. A. D. Holley and W. R. Holley. *Anal. Chem.*, 1952, **24**, 216–18.—Alcohols may be identified in dilute aqueous solution by preparing the 3 : 5-dinitrobenzoate and separating the 3 : 5-dinitrobenzoate chromatographically on a silica–diatomaceous earth column, using specially purified petroleum ether as the eluant. The concentration of the 3 : 5-dinitrobenzoate in each fraction can be estimated spectrophotometrically. J. S.

2141. Determination of nitrogen. Modified Kjeldahl procedure using thiosalicylic acid. P. McCutchan and W. F. Roth. *Anal. Chem.*, 1952, **24**, 369–70.—The macro Kjeldahl procedures generally used for the determination of nitrogen in petroleum fractions do not give quantitative results with samples containing nitrogen in nitro-type compounds. A simple modification of the Kjeldahl procedure permits the rapid determination of nitrogen in such compounds as nitrobenzene and nitromethane. Thiosalicylic acid is used to assist the conversion of the oxidized nitrogen to ammonia. This modification

extends the scope of the method and gives greater assurance of obtaining accurate values on samples of unknown structure. It is eminently suited for control operations, as some twenty determinations can be made per man-day. J. S.

2142. Determination of small amounts of molecular oxygen in gases and liquids. F. R. Brooks, M. Dimbat, R. S. Treseder, and L. Lykken. *Anal. Chem.*, 1952, **24**, 520–4.—In the petroleum and related industries the need frequently arises for a reliable method for determining dissolved oxygen in both aqueous and non-aqueous liquids. A knowledge of the dissolved oxygen content of such liquids as gasoline, kerosine, lubricating oil, solvents, extracting solutions, and treating solutions is often of value in the solution of both plant and laboratory problems involving corrosion, emulsions, and storage stability of refined products. The Mac Hattie–Maconachie method for the determination of traces of oxygen in gases by reaction with copper has been improved and extended to the determination of dissolved oxygen in aqueous and non-aqueous liquids. Dissolved oxygen is determined by stripping the oxygen from solution by a stream of nitrogen, which is then passed through the reaction cell to remove oxygen. Data have been collected which indicate that the molecular oxygen content of both liquid and gas samples can be determined within 5% accuracy over a wide concentration range. J. S.

2143. Determination of rubber hydrocarbon by a modified bromination method. W. J. Gowans and F. E. Clark. *Anal. Chem.*, 1952, **24**, 529–33.—Published investigations show that in the bromination of rubber, addition is accompanied by some substitution. The substitution portion of the bromination reaction is difficult to control, and has contributed to low results and occasional poor precision. This work was carried out to investigate the effect of several factors on the bromination reaction and to find a means of inhibiting the substitution reaction. The effect of rubber hydrocarbon concentration on bromination is critical. Iodine accelerates bromination, but is not present in the final product. Temp and light are important variables. The bromination reaction is insensitive to bromine concentration, so long as a considerable excess is present. Trichloroacetic acid in the amounts used to assist in the solution of crude rubber has no effect on the results. Addition of chloroform to the benzene solution of rubber inhibits the substitution reaction with no apparent effect on the addition reaction. A factor of 0·299 for converting rubber bromide to rubber hydrocarbon is obtained for Hevea smoked sheet, which agrees well with the theoretical value. A conversion factor of 0·301 is obtained for guayule rubber, which may or may not indicate that guayule rubber hydrocarbon is slightly more saturated than Hevea. The method should have applications for the direct and accurate determination of rubber hydrocarbon in crude rubber. J. S.

2144. Determination of microgram quantities of sulphide sulphur. B. L. Dunicz and T. Rosenquist. *Anal. Chem.*, 1952, **24**, 404–6.—A reliable method is needed for the determination of microgram quantities of hydrogen sulphide and sulphur. A volumetric method based on the oxidation of hydrogen sulphide into sulphate by means of alkali hypochlorite has been investigated. Systematic studies show that this reaction is influenced by pH and the temp of the hypochlorite solution. Quantitative oxidation of sulphide into sulphate occurs at room temp in hypochlorite solutions of pH between 8 and 11 and between 14 and 15. In the intermediate pH range the oxidation is incomplete, and colloidal sulphur is formed. The lowest yield, at 0° C and pH 12, corresponds to about 50% oxidation to sulphate. As most favourable for analytical work, a hypochlorite solution of pH 14·4 is suggested. The method can be used for determining sulphur whenever it is present as hydrogen sulphide, or can be converted into it. J. S.

2145. Manometric determination of low concentrations of ethylene. With particular reference to plant material. R. E. Young, H. K. Pratt, and J. B. Biale. *Anal. Chem.*, 1952, **24**, 551–5.—Ethylene gas in minute quantities plays an important role in plant metabolism, especially in the ripening of fruit. A sensitive quantitative analytical method has long been needed. In the method reported, ethylene is absorbed from an air stream in a solution of mercury perchlorate, forming an ethylene mercury complex. The accumulated ethylene is later released from the complex by the addition of hydrochloric acid, and its vol is measured manometrically. This method is best

suited for ethylene of concentration of 0·5 p.p.m. and higher, though not more than 0·05 p.p.m. will escape absorption. Total amounts of ethylene as low as 0·2 ml can be determined with an accuracy of 5%, while a higher accuracy can be obtained with larger amounts. Although the method is specific for olefins only, no gaseous olefin other than ethylene is known to be produced by plants. A number of other volatile products of plant metabolism were tested and did not interfere. J. S.

2146. Determination of cyanides in refinery waste waters. N. Brooke. *Anal. Chem.*, 1952, **24**, 583–4.—A colorimetric method for the determination of cyanides has been developed which does not require preliminary distillation and is sensitive in the range of 0·5 to 3 p.p.m. of cyanide. Although it was developed primarily for use on refinery waste waters containing alkali cyanides, the method could be applied to other problems requiring the determination of trace quantities of cyanides.

Sulphides interfere with the test, and can be removed by treating the neutral or slightly acid waste with cadmium nitrate and filtering. Hydroxide formation of some heavy metals can be suppressed by addition of citric or tartaric acid, but complete removal by ion exchange is preferable. High concentrations of phenols and phosphates interfere. J. S.

2147. Electro-capillary analysis of detergents. W. Kopaczewski. *Chim. et Ind.*, 1952, **67**, 761–6.—Previous work on electro-capillary analysis (procedure analogous to paper chromatography) indicates that addition of electro-negative, electro-positive, or amphoteric colloidal dyes to liquids enables the following properties of micelles to be assessed: electric charge, degree of hydration, variations in surface tension, visc, pH, rH, degree of dispersion and the presence of salts and other impurities. Dyes used should be non-dialyzable and freed from ionic impurities by extensive dialysis, *e.g.*, direct black (−ve), Na alizarine sulphonate (amphoteric), Janus green (+ve). Tests are carried out with 1 cm wide, 20 cm long filter paper strips immersed in 10 cm^3 of test liquid, duration is 20 hr. Control tests are done on same dyes dispersed in dist water. Results show that " anionic " detergents carry +ve charge (*i.e.*, they prevent penetration into paper of −vely charged dyes and facilitate that of +vely charged), " cationic " detergents show inverse effect, and " non-ionic " detergents facilitate dye penetration irrespective of charge of latter. Since electric charge of detergent is considered to be vital factor in its physical-chemical behaviour (dispersion, emulsification, etc.), it is suggested that (for products giving colloidal dispersions) detergent classification should be on this basis rather than on present " chemical " division. Wetting properties of detergents can be assessed by height of rise of dye dispersion on filter paper, all detergents examined (150) decreased this, thus indicating hydrophilic characteristics. Experimental results are illustrated by photographs. V. B.

2148. Evaluation of antibacterial activity. E. M. Britt and L. B. Schweiger. *Soap Sanitary Chem.*, June 1952, **28** (6), 159.—The usual tests to evaluate a new antiseptic are discussed, and the following tests carried out at the authors' laboratory are given in detail: (1) F.D.A. tests for liquid antiseptics; (2) Kolmer bacteriostatic test; (3) percentage-kill test; (4) filter paper disk method; (5) a practical-use method involving hand-washing experiments. N. W. G.

Crude Oil

2149. Chemical state of vanadium in Santa Maria Valley crude oil. D. A. Skinner. *Industr. Engng Chem.*, May 1952, **44** (5), 1159–65.—Vanadium was found to exist in the above crude as a vanadium–porphyrin complex largely in the asphaltic fraction. Its stability and water insolubility substantiate the marine sediment origin of petroleum. Knowledge of the chemical state of vanadium in petroleum should be helpful in investigating problems associated with its presence. E. J. C.

Engine Fuels

2150. Alcohol as an anti-knock agent in automotive engines. J. C. Porter and R. Wiebe. *Industr. Engng Chem.*, May 1952, **44** (5), 1098–104.—Knock rating values of blends of ethyl alcohol with various gasolines are reported in tabular and graphical form. Blends of *iso*-octane and *n*-heptane, and toluene and di-*iso*butylene gave

highest octane gain with alcohol, and lowest gain was obtained with a highly cracked gasoline. Octane gains of alcohol and T.E.L. were found to be largely independent up to 25% C_2H_5OH and 3 ml T.E.L/U.S.G. Presence of T.E.L. had no significant effect on octane blending value of alcohol, except in highly cracked fuels, where alcohol gave greater gain in absence of lead. S did not appear to affect gain of alcohol as with lead. E. J. C.

2151. Liquid methane as a motor fuel. M. Pearce. *Proc. Instn Mech. Engrs* (*Auto. Div.*), 1949–50, Pt III, 155–66.—Details are presented of an investigation of the use of liquid methane as a motor fuel started in 1938 in the Dept. of Chemical Engineering and Applied Chemistry, Imperial College. The properties of methane are tabulated, and its effectiveness as a motor fuel, indicated by comparative tests at the Ricardo laboratories, is discussed. The vehicle fuel system and its components as finally developed is described, together with the results of long-distance service trials with a motor bus with normal and higher compression ratio. Loss of fuel by evaporation constitutes a present disadvantage; modifications in the fuel system are suggested for its elimination. J. G. H.

2152. Oxidation decompostion, ignition, and detonation of fuel vapours and gases. Pt XIX. Optimum timing of compression or spark ignition as determined by adjustment of compression ratio; acetaldehyde and diethyl ether as engine fuels. R. O. King, E. J. Durand, A. B. Allen, and E. J. T. Hanson. *Canad. J. Tech.*, 1952, **30**, 29–43.—A continuation of the work reported in Pt XVIII of this paper (Abs. No. 1132).

Acetaldehyde and diethyl ether were used as the fuels for a CFR carburettor-type engine run in the cool condition required to ensure the presence in the charge of liquid drops during compression, vaporization of the fuel in the carburettor being restricted by maintaining the air supply at the relatively low temperature of 50° C, and by removing the diffuser. Charge density was subnormal, and C.R. was then always adjustable to the value required for max power output (the optimum value) without inducing knock of intolerable intensity when using rich mixtures. The spark plugs of the engine were replaced by blind plugs for experiments with compression ignition, and both fuels were used with mixture strengths varying from 75% weak, B.H.P. then being zero, to 140% rich. The graphs relating optimum C.R. to mixture strength are of a W form with two min values of optimum C.R. and an intermediate max value. Spark ignition was not effective with mixtures leaner than 40% weak or more than 80% rich. A relatively low engine speed of 400 r.p.m., and the presence of liquid drops in the charge during compression, provided conditions suitable for liquid-phase cracking, and the compression ignition obtained in the conditions of the experiments is attributed to the effect of nuclei formed accordingly. The experimental results afford evidence of thermal efficiency being affected adversely by the loss during compression of the heat required by the endothermic cracking reaction which produces the nuclei required for compression ignition. S. B.

2153. Oxidation decomposition, ignition, and detonation of fuel vapours and gases. Pt XX. The adverse effect on thermal efficiency of the endothermic reaction required for nuclear ignition as shown by engine experiments with methanol, acetaldehyde, and diethyl ether. R. O. King and A. B. Allen. *Canad. J. Tech.*, 1952, **30**, 44–60.—In Pt IV of this paper (Abs. No. 1690—1948) a nuclear theory of ignition was stated. Auto-ignition in engines may, depending on C.R. and the nature of the fuel, occur before, after, or without spark ignition, and is due, according to this nuclear theory, to the heat of oxidation reactions on material articles formed by the cracking of the fuel. Cracking is an endothermic reaction, and the consequent absorption of heat in the gaseous phase would be expected to result in a lower thermal efficiency than would otherwise be obtained. In this part of the paper the results of experiments with methanol, acetaldehyde, and ether have been used in an endeavour to verify the validity of the theory. Methanol cannot be decomposed in the engine to the extent necessary for the formation of material nuclei, and therefore auto-ignition was not obtained. Acetaldehyde and diethyl ether, however, decompose relatively easily, and auto-ignition therefore occurred at relatively low C.R.s.

Corresponding thermal efficiencies are exhibited by a series of graphs which show the extensive deviations from ideal efficiencies observed when acetaldehyde and diethyl ether were used as fuels compared with that observed when methanol was used. The

deviations are as would be expected taking heat of activation into account, and it is indicated that in most cases a pressure wave is required for the nuclear ignition of other than a small volume of unburned mixture remaining late in the period of combustion when the ignition is by spark. The experiments described in this part are in respect of an hitherto unexplored aspect of the nuclear theory of ignition, and are to be regarded as preliminary to more detailed ones with hydrocarbon fuels. S. B.

Gas Oil and Fuel Oil

2154. Oil gas manufacture. W. Q. Hull and W. A. Kohlhoff. *Industr. Engng Chem.*, May 1952, **44** (5), 936–48.—A U.S. plant for manufacturing domestic gas from oil is described with the aid of flowsheets. Details are given of gas production, gas treatment, by-product recovery, and refining. A further plant for producing petroleum coke for electrode manufacture is also described, and this yields a gas of high calorific value (1030 B.Th.U/10^3 cu. ft.) Research which has been carried out on high B.Th.U. gas is also reported. E. J. C.

2155. Patent. U.S.P. 2,582,192 (13.7.48; 8.1.52). G. H. Denison, assr to California Research Corporation. Diesel fuel oil containing a homologous mixture of heterocyclic secondary amines prepared by extracting a crude oil under specified conditions with dilute mineral acid and catalytically reducing the extracted bases to inhibit gum formation. R. C. R.

Lubricants

2156. Review of the BSI draft specification for engine lubricating oils. (H.D. type). Anon. *Scientific Lubrication*, Mar. 1952, **4** (3), 28–9.—The shortcomings of the new Ministry of Defence specification DEF/2101 as a standard for commercial requirements are indicated, and the correlations between the new BSI draft specification DEF/2101 and the U.S. specification MIL-O-2104 are pointed out. The absence of reference to " re-branding " in the draft is noted, as is the lack of equivalent for the old Supplement 1 oils and for lubricants for two-stroke C.I. engines. J. G. H.

2157. Effects of light on lubricants and fuels. R. K. Ross. *Scientific Lubrication*, Mar. 1952, **4** (3), 12–15, 27.—The fundamentals of colour and colour measurement are enumerated, and the Lovibond Tintometer, ASTM Union Colorimeter, the Saybolt Chromometer, and the Tag-Robinson Colorimeter are briefly described. The Colour Index as obtained by the ASTM " Proposed method for determination of Color Index of petroleum products by photoelectric colorimeter " is outlined, and correlations of Colour Index with results obtained by the methods previously described are presented in tabular form. J. G. H.

2158. Use of infra-red radiant heat equipment in the lubrication industry. E. S. Morley. *Scientific Lubrication*, Jan. 1952, **4** (1), 23–5.—The principles and methods of application of infra-red radiant heat are outlined, and its employment in barrel stoving ovens, the extraction of grease from open-ended steel barrels and in oil barrel draining, and the advantages accruing from the use of infra-red equipment are discussed. J. G. H.

2159. Proposed list of mathematical symbols for lubrication problems. J. Boyd. *Lubr. Engng*, Dec. 1951, **7** (6), 282–3.—A tentative list of mathematical symbols put forward by the ASLE with a discussion of general principles and comments on individual symbols. J. G. H.

2160. More lubrication problems from overseas. R. W. Tong. *Scientific Lubrication*, Jan. 1952, **4** (1), 20–2.—Points treated include cleaning facilities and defects in heat exchangers and oil coolers, screw threads, water and oil cocks, big end troubles, lubrication of valve rocker gear, and problems associated with oil containers. J. G. H.

2161. Automotive drawing compounds—specifications and applications. J. T. O'Reilly. *Lubr. Engng*, Feb. 1952, **8** (1), 14–17.—The dependence on shop tests in the quality control of drawing compounds is remarked and desirable properties enumerated.

Typical material and process specifications are presented, and points in correct application stressed, as well as the importance of selecting the drawing compound with due regard to the conditions encountered. J. G. H.

2162. Some factors governing the performance of crankcase lubricating oils. A. Towele. *Proc. Instn Mech. Engrs (Auto Div.)*, 1949–50, Pt II, 88–111.—The basic requirements of a crankcase oil are enumerated, and the chemical and physical tests commonly carried out on new and used oils are noted. The laboratory methods for determining visc and the connexion with SAE Nos. and V.I. are discussed, together with practical problems encountered in lubrication. Ring sticking, engine wear, bearing failures, and engine deposits are considered, together with the influence of fuels on engine deposits and wear, engine laboratory tests, and the correlation of field and laboratory tests. It is suggested that many features of unsatisfactory engine operation often attributed to the lubricant are occasioned by incorrect design, excessive operating conditions, or poor maintenance. The value of additive-treated oils is indicated, and the value noted of Supplement 1 oils when high sulphur fuel is used. J. G. H.

2163. Metal transfer from piston rings to cylinders during " run-in." C. D. Strang and J. T. Burwell. *Proc. Instn Mech. Engrs (Auto. Div.)*, 1949–50, Pt IV, 169–75.—Details are presented of an investigation of the micro-welding between rings and cyl wall during " run-in " in a small water-cooled I.C. engine, using piston rings with radio-active wearing surfaces. Results obtained indicated that micro-welding and the accompanying transfer of metal occurred under the mildest conditions of engine operation, including motoring with the engine cyl head removed. Distribution of metal along the ring travel corresponded to the wear profile reported by other workers; it is suggested that such transferred metal may play a part in the formation of the " glazed " layer said to be present on " run-in " cyl walls and rings. The occurrence of transferred metal at all points along the ring travel indicates that the piston ring was not fully supported by a hydraulic oil film of thickness sufficient to separate completely the surfaces of the ring and the cyl wall. Procedure for the preparation of the plating solution from radio-active chromium metal is outlined. J. G. H.

2164. Centrifuging of used oils. A. C. Smith. *Scientific Lubrication*, Dec. 1951, **3** (12), 15–19.—The widespread use of centrifuging in the reconditioning of used lubricating oil is remarked, and the principles of centrifuging are briefly described, together with the design and operation of centrifuges, classification, and separation. Methods of ring dam adjustment for efficient separation are discussed in some detail, as are water washing, capacity and throughput, optimum temp for centrifuging, and the effect of centrifuging on detergent oils. J. G. H.

2165. Can H.D. oils be filtered? A. V. Driver. *Scientific Lubrication*, Dec. 1951, **3** (12), 24–5.—The possible sources of contamination of H.D. oils are enumerated, and the necessity for filtration is emphasized. The available types of filter are detailed and their characteristics indicated. The use of activated earths and chemical coagulants in the reclamation of H.D. oils is to be avoided, and the use of centrifuges is only partially effective. Additive depletion is discussed, and mechanical means of filtering are considered to be the only satisfactory procedure. J. G. H.

2166. Filtration of industrial and lubricating oils. J. R. McCoy. *Lubr. Engng*, Apr. 1952, **8** (2), 66–73.—The value of continuous filtration in the maintenance of clean oil is stressed, and the necessity of employing special-type filters for specific purposes is indicated. Depth-type, bulk-type, inert, edge-type, and magnetic filters are described in some detail. The appropriate filtration equipment for I.C. engines, cutting oils, quench oils, hydraulic and insulating oils is detailed, and special points attendant on the filtration of synthetic fluids and lubricants are indicated. J. G. H.

2167. Animal and vegetable oils, fats, and waxes. C. Griffiths. *Scientific Lubrication*, Apr. 1952, **4** (4), 10–14; May 1952, **4** (5), 15.—In Pt 1 the classification and chemical composition of animal and vegetable oils, fats, and waxes are discussed, and the special properties imparted by " blowing " are indicated. The advantages of sulphurizing

and sulphonating are touched upon, and the specific applications of drying and semi-drying vegetable oils in other than lubricating fields are enumerated. The functions of castor and rape oils in lubrication are considered in some detail.

In Pt 2 the properties and fields of application of neatsfoot, lard, fish, whale, and seal oils are outlined, together with similar treatment of vegetable and animal fats and waxes. J. G. H.

2168. Fatty oils in lubrication. K. G. Slim. *Scientific Lubrication*, Dec. 1951, **3** (12), 26.—The occurrence, refining, and constitution of fatty oils are discussed and the properties of significance in lubrication indicated. The application of rape, fish, and castor oils are noted. Fatty acids and saponification are considered, and the occasions when fatty acids can be effectively used are enumerated. J. G. H.

2169. Synthetic oils for auto and steam cylinders. H. Kubbel and A. Meusel. *Scientific Lubrication*, Apr. 1952, **4** (4), 19.—Chemische Werke Rheinprussen synthesized lubricating oils during the war period by alkylating naphthalene. Production, used for lubricating turbines, steam cyls, and I.C. engines, is stated to have reached 3600 tons/annum during 1942–45. Details are presented of the automobile engine and steam turbine, together with the procedures, used for test purposes. J. G. H.

2170. Lubrication of knitting machinery. F. K. Fogleman. *Lubr. Engng*, Apr. 1952, **8** (2), 74–5, 101.—Recent tendencies in the lubrication of knitting machinery tend away from speciality lubrication to the use of four types only. Typical knitting machinery of the " Reading " type is described, lubrication points being discussed in some detail, with a stainless needle oil carrying most of the burden. Emphasis is laid on the quality of personnel engaged in lubrication duties. J. G. H.

2171. Refrigerator oils. B. J. Vaughan. *Scientific Lubrication*, Jan. 1952, **4** (1), 26–8.—The performance and properties desirable for a low temp lubricant are discussed, a solvent refined oil, preferably naphthenic base with a V.I. of 75 to 85, being recommended, with a setting point of between −20° and 40° F. Methods of determining pour point are outlined and details presented of the " flocculation test." Chemical stability is considered from the standpoints of stability towards oxidation and polymerization, effect of moisture, copper plating when halogen refrigerants are used, and laboratory assessment of the oil. Methods are indicated for the estimation of moisture. J. G. H.

2172. Patents. U.S.P. 2,580,430 (24.8.46; 1.1.52). E. C. Hughes and J. D. Bartleson, assrs to The Standard Oil Co. Lubricants produced by reacting a saturated hydrocarbon oil with 1 to 20% of a phosphorus sulphide in the presence of a halide catalyst.

U.S.P. 2,580,570 (24.4.50, 1.1.52). P. R. McCarthy, assr to Gulf Research & Development Co. A smooth-textured lithium base grease comprising mineral oil, a lithium soap, and a hydrogenated rosin of at least 40% saturation.

U.S.P. 2,580,587 (14.11.49; 1.1.52). F. P. Otto and R. W. Barrett, assrs to Socony-Vacuum Oil Co. Lub oil containing a pour-point depressant consisting of a polymeric *n*-tetradecyl metaphosphimate obtained by reacting a phosphonitrillic halide with *n*-tetradecanol and polymerizing the product.

U.S.P. 2,580,654 (26.10.49; 1.1.52). G. V. Browning, assr to Standard Oil Co. An aerosol dispensable lubricant consisting of a lithium soap–oil grease having a soap content of 4 to 20% dissolved in a mixture of trichloromonofluoromethane and dichlorodifluoromethane, the oil component of the grease also being dissolved therein.

U.S.P. 2,581,126 (31.8.49; 1.1.52). A. J. Morway and J. J. Kolfenbach, assrs to Standard Oil Development Co. Extreme pressure lub grease comprising a lub oil thickened with a combination of a metal soap of higher fatty acids and the non-volatile Cannizzaro reaction products of an alkali metal hydroxide and a sulphur-containing heterocyclic aldehyde.

U.S.P. 2,581,127 (25.1.50; 1.1.52). A. J. Morway and J. J. Kolfenbach, assrs to Standard Oil Development Co. Lub grease comprising a lub oil thickened with a metal salt of an aliphatic ketonic acid and a metal soap of a fatty acid.

U.S.P. 2,581,132 (13.7.49; 1.1.52). R. F. Nelson and M. R. Hefty, assrs to the Texas Co. Lub oil containing a substituted glyoxalidine and adapted to form a water-in-oil emulsion in the presence of water.

U.S.P. 2,581,626 (6.10.48; 8.1.52). J. W. Brooks, assr to Socony-Vacuum Oil Co. Mineral oil containing up to 10% of a thienylthio carboxylic acid or ester as an anti-oxidant.

U.S.P. 2,582,708 (1.10.48; 15.1.52). S. B. Lippincott and A. H. Popkin, assrs to Standard Oil Development Co. Lub oil additives comprising esterification condensates obtained by reacting ethylene—1 : 2-dicarboxylic acids or methylene succinic acids or their anhydrides with butadiene monoxide and a straight chain primary alcohol containing 8 to 24 C atoms as esterification agent.

U.S.P. 2,582,958 (16.10.48; 22.1.52). E. W. Brennan and N. D. Williams, assrs to The Pure Oil Co. Lub oil additive comprising a metal salt of a sulphurized and phosphorized wool grease. R. C. R.

2173. Abrasion and wear. Anon. *India-Rubber J.*, 1951, **121**, 840.—A description is given of the International Symposium held on 14/15 November at the new laboratories of the Rubber-Stichting, Delft. Summaries are included of the papers presented, three of which are of petroleum interest: (1) "The friction and surface damage of non-metallic solids," by Dr F. P. Bowden. (2) "War on wear," by Prof H. Blok, and (3) "Chemical causes of wear," by Prof J. J. Breoze. C. N. T.

Bitumen, Asphalt, and Tar

2174. Chemistry and rheology of asphaltic bitumen. Pt II. S. L. Neppe. *Refiner*, 1952, **31** (4), 184.—See Abs. No. 1704. The effect of chemical composition on the rheology of asphalts is discussed. The importance of temp susceptibility is stressed, and the problems of cracked bitumens and heterogeneity as related to the Oliensis test are described. Restrictions in certain bitumen specifications are criticized as being unjustified, and the necessity for further research is emphasized. (Author's Abstract.) A. R. H.

2175. The flowing of bitumen. H. Nüssel. *Bitumen, Teere, Asphalte, Peche*, 1952, **1** (3), 1–5.—A procedure is presented for determination of the tendency of bitumen to flow under the influence of temp, time, and angle of inclination. The apparatus designed by the Dutch testing office for testing bituminous coatings on iron pipes, on which the angle of inclination is adjustable, is illustrated as used. Tabulated results and curves obtained for flat moulded specimens of twelve bitumens show the effects of ext influences, thickness of layer, angle of inclination, and time. Int factor is soft. pt. reflected in the pen index. The soft. pt. gives a reference pt but no accurate measure for the inclination to flow. From the results precise conclusions may be drawn that the soft pt. of adhesive masses for composition roofs must lie sufficiently high above the max temp occurring on the roof. Foreign material, tar 2% or tar oil 5%, suffices to double the flow length. Such fluxed bitumens flow much more strongly than pure bitumen of the same soft. pt. R. T.

2176. Relationship between compression strength of a cube and optimum binder addition. G. Herion. *Bitumen, Teere, Asphalte, Peche*, 1952, **1** (3), 5–11.—Determination of the dependence of the compression strength of a cube on the amount of binder added. Special cubes prepared to DIN1996U54 using the drop hammer illustrated were tested. The results are reproduced in curves for a series of mineral mixtures with bitumen, tar, and tar-bitumen. With a low binder visc estimation of optimum binder cont from max compression strength of cubes is limited by the decreasing differences in compression strength with change of binder addition. If the increase or decrease in the val falls below the permissible test inaccuracy of $\pm 10\%$ by $\pm 0{\cdot}5\%$ the lower permissible limit for practice in the estimation of the binder cont according to the max compression strength is reached. Mineral mixtures with varying quantities of different binders give sufficiently dependable results only within the above limits. Reliable estimation of optimum cont with increasing stiffening of the binder is possible only when Einstein's visc law applies. This investigation range, varying with type of

binder and visc, is limited with increasing visc by the crit. visc. If the crit. visc. be exceeded, a compression strength max is obtained with a binder addition exceeding the optimum proportion. This applies also to mineral mixtures with different binders. Binder cont corresponding to max compression strength is dependent both on binder visc and binder and mineral characteristics. R. T.

2177. Patents. U.S.P. 2,581,685 (16.6.48; 8.1.52). W. A. McMillan, assr to The Texas Co. In the manufacture of asphalt roofing felt, the use of a small amount of a liquid dihydrocarbon silicone polymer to inhibit foaming of the molten asphalt bath.

U.S.P. 2,582,823 (23.3.48; 15.1.52). F. M. Fowkes, assr to Shell Development Co. Asphalt composition comprising an asphaltic material, a solid, and an amine salt of a water-soluble acid-acting agent.

U.S.P 2,582,824 (23.3.48; 15.1.52). F. M. Fowkes, assr to Shell Development Co. Bituminous composition comprising a wet aggregate, an asphalt, and a bonding agent consisting of an amine salt of a water-soluble polyamine and a fatty acid of at least 12 C atoms. R. C. R.

Special Hydrocarbon Products

2178. Composition of naphtha from fluid catalytic cracking. F. W. Melpolder, R. A. Brown, W. S. Young, and C. E. Headington. *Industr. Engng Chem.*, May 1952, **44** (5), 1142–6.—Mass, ultra-violet, and infra-red spectroscopy, fractional distillation, adsorption, and hydrogenation were used to isolate chemical constitutents of a 94° to 424° F boiling range fluid cat cracked naphtha. Ninety-two hydrocarbons, including twenty-one olefins and sixty groups or classes of compound, were determined. E. J. C.

Coal, Shale, and Peat

2179. Shale oil production and refining today. Pt III. P. W. Sherwood. *Refiner*, May 1952, **31** (5), 160–4.—The production of gasolines and diesel oil from shale oil is discussed, including cracking, coking, and hydrogenation. The economics of gasoline production from oil shale are also discussed. A. R. H.

2180. Metropolitan [French] oil shales. V. Charrin. *Chim. et Ind.*, 1952, **67**, 825–31.—Brief historical review of French shale oil industry and of geological distribution of shale in France is followed by short notes on each of the fourteen main French deposits of oil shale. Working is now carried out only at Autun, where it has been continuous since 1835; present daily plant capacity is 1000 tons shale. It is considered that national shale resources have been neglected and that remunerative operation is quite feasible. V. B.

2181. Studies of low-grade Indian coals. Pt 2. Experimental low-temperature carbonization and reactivity of the low-temperature coke. S. S. Ghosh *et al. Brennstoff-Chem.*, 1952, **33** (3–4), 47–55.—Low-temp carbonization at 500° C of seven low-grade coals from the Iharia field and two from the Ranigunj field discussed in Pt 1 is dealt with. The reactivity of two low-temp cokes therefrom is discussed with regard to large-scale use of the coals for low-temp carbonization, hydrogenation, or complete gasification. Plasticity and swelling, observed in most cases, make low-temp carbonization difficult. Microscopic structures of low-temp cokes support this viewpoint. The structural peculiarities of the cokes bear a relationship to the CH_4/H_2 in the gas and light tar fractions. The plastic tar seam developed influences the progress of low-temp carbonization and the nature of the products. The reactivity of two representative coke samples was compared with that of a typical foundry coke. The plasticity of the coal likewise influenced the reactivity of the coke. Coke from a highly carbonized plastic coal is less reactive than that from a less carbonized coal. As a generalization, conditions favouring a high yield of light tar fractions and development of a fine micellar structure in the coke induce graphitization and reduce reactivity. Plasticity development in bituminous coals depends on degree of carbonization, but a limit to carbonization age of the coal affects the degree of plasticity. More highly

carbonized coals, as samples 8 and 9, give a less viscous tar seam and coke of uniform cellular structure. Samples 1 and 2 are most suitable for low-temp carbonization with a max tar yield, due to their low plasticity. These less carbonized coals are suitable for hydrogenation. Coals of the Ihara field are unsuitable for low-temp carbonization or hydrogenation, but are suitable for complete gasification. R. T.

2182. Scottish shale oil. A review of an important British industry—II. Anon. *Petroleum*, June 1952, **15** (6), 158–60.—Pt **2** of a review of the Scottish shale-oil industry (see also Abs. 1716). Main processes carried out at the Pumpherston refinery in the refining of crude shale oil to produce motor spirit, solvent naphthas, diesel oil, paraffin wax, and synthetic detergents are described. R. E. P.

Miscellaneous Products

2183. Alkyl benzene sulphonates as detergents. A. Davidsohn. *Industr. Chem.*, 1952, **28**, 198–203.—Principal methods for production of dodecyl benzene are described in detail; reaction of benzene with chlorinated pet fraction under Friedel–Kraft conditions or sulphuric acid catalysed alkylation with polymerized olefins. Sulphonation, acid neutralization processes are discussed and importance of temp control and exclusion of iron emphasized, and a summary of conclusions reached on spray drying of Santomerse No. 1 given. Some formulations for dishwashing and laundry detergents are tabulated, and a new detergent based on alkyl benzene sulphonates is described which contains organic solvent and is soluble in water and organic liquids. The solvent-containing detergent is stable in hot 2 to 3% H_2SO_4 and resistant to alkalis. E. B.

2184. Detergent measurement. E. H. Armbruster and G. M. Ridenour. *Soap Sanitary Chem.*, June 1952, **28** (6), 83.—The older methods, carbon black removal and the use of fluorescent dyes are compared with a new method employing radio isotopes. The advantages claimed for the new method are greater sensitivity so that lower build-up can be determined, and the results are independent of the physical nature of the test surface. Two examples are given using Phosphorous 32 as a tracer element—the removal of bacteria from cloth in self-service laundries and the washing of cutlery in large eating establishments. N. W. G.

2185. Silicones—new materials and possibilities for their use. W. Chajec. *Nafta*, 1951, **7**, 87–90 (from "Silicone Chemistry," by Nyul Gyula, *Magyar Kemikusok Lapja*, Nos. 3 and 6, 1950).—Brief description of structure and properties of silicon oils, plastics, resins, and greases which contain Si, O, H, and often hydrocarbon groups. These materials are found to be much superior to their organic analogues and occasionally show quite new properties. M. S.

2186. Cationic surface active agents. V. Dvorkovitz. *Soap Sanitary Chem.*, May 1952, **28** (5), 41.—This article discusses the general principles of mol structure of surface-active agents, with particular reference to amines as the principle cationics. The main uses are grouped: (*a*) for the textile industry as softening agents due to the ease of absorption of cationics on fibres, as dye assists, and in production of water repellent finishes; (*b*) as a flotation agent; (*c*) as a micro-biocide—newer work on cationics has shown that resistant organisms are not killed but the cationic adsorbed and acts as a bacteriostat; (*d*) as an antiseptic, with particular discussion on applications in the dairy industry. N. W. G.

2187. Spray drying. Anon. *Soap Sanitary Chem.*, May 1952, **28** (5), 67.—A summary of an article by M. Fock in *Alchemist*, Mar. 1952, illustrating the structure of spray drying plants using both nozzle and disk methods for soap and detergent spraying; the ease of resolution is given by the hollow particles resulting from these plants. With high solid contents no air heating can be practised, the solidification being given by crystallization, but where appreciable amounts of water have to be removed, heated air is supplied to the drying chamber. Included in most systems are supplementary bag filters to recover fine particles carried out of the chamber by the air stream. N. W. G.

2188. Modern floor polishes. M. A. Lesser. *Soap Sanitary Chem.*, May 1952, **28** (5), 107.—A review of wax polishes with particular emphasis on self-polishing types, since approximately 80% of the marketed products in the U.S. are self-polishing. Representative formulæ from the literature—mainly from patents—are included for paste and liquid floor waxes, and replacements for carnauba wax are discussed. The main commercial products available are other vegetable waxes, such as candelilla and oxidized microcrystalline paraffin wax. The modern developments using silicones as additives are also covered. N. W. G.

2189. Action of germicides. S. Boyk. *Soap Sanitary Chem.*, May 1952, **28** (5), 111.—A theory is advanced to explain the action of a germicide in killing of micro-organisms and the resistance of the system to various additives. It is suggested that the lethal action is due to the adsorption of the germicide by the protein of the microbe, followed by irreversible reaction in which the protein is permanently altered, resulting in the death of the microbe. This idea is used to explain the variation in phenol coefficient for chemical series of products and for the effects produced by hydroxyl ions, salt, solvents, soaps, and neutralizing agents. N. W. G.

2190. Pyrethrum synergists. R. L. Blickle. *Soap Sanitary Chem.*, May 1952, **28** (5), 121.—Tests using houseflies were carried out using control and synergized pyrethrum sprays, with the standard spray giving a mortality figure in the range 20 to 30%. The greatest increase was given with diethyl 3 : 6-endomethylene-4-tetrahydrophthalate, which increased the mortality by 51·4% when used as a 5% solution in kerosine with the standard spray. Sixty-six materials were tested with the results shown in table form. N. W. G.

2191. Collapsible rubber drums. Anon. *India-Rubber J.*, 1951, **121**, 21.—According to the May 1951 issue of "US," the United States Rubber Co.'s newest development consists of rubber fabric barrels or drums for the shipment of liquids. The drums can be collapsed after emptying and are re-usable; 2500 collapsed drums occupy the space of 300 rigid ones. They contain about 36 I.G., and tests have indicated that the new drums will be suitable for oils, greases, fats, acids, etc., and may be suitable for dropping supplies from the air. C. N. T.

2192. A study of plasticized high Mooney viscosity synthetic rubbers. G. H. Swart, E. S. Pfan, and K. V. Weinstock. *India-Rubber World*, 1951, **124**, 309.—The work described was started to investigate the possibility of plasticizing very high Mooney visc synthetic rubbers of the butadiene copolymer type (previously considered unusable) with inexpensive softeners, such as petroleum oils, to give low-cost rubber compounds for use in mechanical goods. The details of the process as finally developed are disclosed in this paper. A comparison is given of the properties of the various petroleum oils used in the experiments. C. N. T.

2193. GR-SX and the history of co-polymerization. W. H. Stevens. *India-Rubber J.*, 1951, **121**, 465.—A historical review is given of the processes of extending and plasticizing GR-S by means of a substantial addition of mineral oil. C. N. T.

2194. The reinforcement of rubber by carbon black. D. Parkinson. *Brit. J. Appl. Phys.*, 1951, **2**, 273.—Recent research on the carbon reinforcement of rubber is reviewed and existing theories examined. The stiffening effect of carbon black is thought to be little influenced by the size of the individual spherical particles, but values of tensile strength, tear strength, and abrasion resistance increase with decreasing particle size of the black.

Departures from this generalization are considered and possible factors discussed. The nature of the attachments between carbon particles and rubber molecules forms the subject of much current research, the results of some of which indicate the possibility that chemical cross linkages supplement physical adsorption at the carbon-rubber interface. (Author's Abstract.) C. N. T.

2195. Production and utilization of high Mooney viscosity synthetic rubber. H. D. Harrington, K. V. Weinstock, N. R. Legge, and E. B. Storey. *India-Rubber World*, 1951, **124**, 435; 571.—A description is given in Pt I of the further development of oil-

plasticized copolymers of high Mooney viscosity through pilot-plant and commercial production to factory and service trials.

In Pt II factory development and compounding variables are outlined; the results of tyre tests, together with physical properties, processing data, and compounding variables are tabulated for the petroleum oil/polymer blends made by the latex-compounding technique.

It is concluded that, although some problems remain to be solved, the highly plasticized polymer may be readily processed in the factory. Comparisons are given with other types of rubber. C. N. T.

2196. Reinforcing agents and reinforcement. Pt 2. Non-black reinforcing fillers. Pt 3. Reinforcement theory. J. R. Scott. *India-Rubber J.*, 1951, **121**, 384; 424.—Pt 2 deals with reinforcing agents other than carbon black, *e.g.*, the so-called "white-carbon."

Pt 3 defines "reinforcement," and describes the influence of particle size, area, and character of the particle surface, especially with respect to adsorbed gases and the presence of reactive groups on the filler surface. Work on rubber/filler affinity is discussed, and the effect of the crystallinity factor on reinforcement is outlined, all the evidence collected indicating that graphitization impairs reinforcement. C. N. T.

2197. Reinforcing agents and reinforcement. J. R. Scott. *India-Rubber J.*, 1951, **121**, 478; 513.—Topics discussed include the structure of the filler (mainly carbon black), primary structure stability, and measurement techniques such as electron microscopy, oil absorption by the dry black, and shrinkage of an unvulcanized rubber/carbon black stock. An outline is given of the work done on the mechanism of secondary structure formation. Information is summarized on the way in which rubber fillers improve those properties regarded as criteria of reinforcement. Topics discussed include the mechanism of reinforcement, the modulus of vulcanized rubber, tension and tear strengths, and abrasion resistance. An outline is given of the methods used in American laboratories for electron microscope studies of carbon black dispersed in vulcanized rubber. C. N. T.

2198. Modern automobile polishes. M. A. Lesser. *Soap Sanitary Chem.*, June 1952, **28** (6), 144.—Literature on car polishes is reviewed and discussed under five types: (1) *Emulsified oil products for application after car washing*—they give a film of oil with a quick shine but poor lasting; (2) *Cleaner–polisher types*—a cleaner such as a mild abrasive to remove dirt and chalked pigments is added and occasionally a small amount of wax; (3) *Cleaner-polish–wax type*—larger quantities of wax give a cleaning polish with a longer surface life; (4) *pre-wax cleaners* contain solvents and surface active agents to give a dry oil-free surface for the polishing wax; (5) *wax polishes*—in addition to the older paste-type polishes containing blends of vegetable and paraffin waxes, polishes containing silicones and blends for propellent spray application are now widely used. N. W. G.

2199. New look at floor cleaners. L. E. Wells. *Soap Sanitary Chem.*, June 1952, **28** (6), 42.—Discusses the influence of soaps and synthetic detergents on speciality flooring materials such as asphalt, plastic and rubber tiles, linoleum, marble, and terrazzo. The samples were tested by complete immersion for two weeks or surface contact. With linoleum, soaps (having an alkaline reaction) give considerable damage and synthetic detergents very little. With the various speciality tiles, most synthetic detergents were satisfactory, but considerable care in the use of alkaline builders is needed. N. W. G.

2200. Asphalt tile maintenance. J. O. Heppes. *Soap Sanitary Chem.*, June 1952, **28** (6), 151.—The tile is composed of asbestos fibres, mineral pigments, inert fillers, with a binder of resin for light colours and gilsonite asphalt for dark colours.

Any damage during cleaning will be caused by a reaction between the binder and the cleaning material. The Chem. Speciality Manf. Assoc. have issued specifications for soap type, non-soap type, and synthetic detergent cleaners. For surfacing after cleaning, wax finishes are discussed. The only safe type of material to use is a water emulsion wax applied sparingly. The wax can also be added combined with sawdust for sweeping compounds. N. W. G.

2201. Factors affecting the service of leather in wool processing. R. M. Koppenhoefer. *Amer. Dyestuff Rept.*, 1951, **40**, 469.—The type and amount of wool oil used in processing wool have important effects on the life of textile leather. An experimental method is described in which these effects can be studied. The results closely parallel those of mill practice. An explanation is given of how oils of different kinds behave towards leather and how excessive deteriorative effects on leather can be avoided. Some oils can accelerate rate of deterioration because of a tanning-like action exhibited by fatty oil sulphates or mineral oil sulphonates, which are widely used as emulsifiers, and it is concluded that the concentrations of these should be limited to safe levels.

C. N. T.

2202. Pyrethrum at the cross roads. H. A. Smith. *Soap Sanitary Chem.*, June 1952, **28** (6), 155.—During recent years pyrethrum prices have been high and synthetic insecticides—particularly allethrin and chlorinated compounds—have become available. Article reviews the prospects for the pyrethrum industry. On the production side ramularia disease is spreading, high yield strains of flowers are now available, but give difficulties in analysis. Transit losses are high for the better-quality flowers. Technically allethrin is not a synthetic pyrethrum, but should be regarded as a competitive insecticide. The main disadvantage of pyrethrum in comparison with the newer insecticides is that it is a non-residual type, but is still the best contact killer and is non-toxic. The main marketing competition is now with the aerosols. It is considered that the demand for pyrethrum will continue, but the volume of its use will be restricted due to price and lack of residual effectiveness. The former factor is a matter for the growers.

N. W. G.

2203. Patents. U.S.P. 2,580,881 (21.5.49; 1.1.52). C. B. Biswell, assr to E.I. du Pont de Nemours & Co. Petroleum product stabilized with up to 2% of a carboxylic acid salt of a 1-orthohydroxy-arylidene aminoguanidine.

U.S.P. 2,580,996 (16.7.48; 1.1.52). J. M. Butler, assr to Monsanto Chemical Co. Coating composition comprising a hydrocarbon wax and an alkylated polystyrene made by reacting polystyrene with a propylene polymer.

U.S.P. 2,581,092 (2.1.48; 1.1.52). J. D. Garber and W. C. Hollyday, Jr., assrs to Standard Oil Development Co. An oil-soluble condensation product of a copolymer of octadecene and butadiene condensed with an excess of tetradecyl mercaptan.

U.S.P. 2,581,120 (5.5.48; 1.1.52). A. H. Matuszak, assr to Standard Oil Development Co. A rust-inhibiting composition comprising a partial ester of a nitroalcohol in a hydrocarbon solvent.

U.S.P. 2,581,174 (13.4.45; 1.1.52). R. Cross and M. C. Cross. An insecticidal composition comprising 1 : 1-bis (*p*-fluorophenyl) 2 : 2 : 2-trichlorethane in a petroleum hydrocarbon solvent.

U.S.P. 2,581,441 (26.2.48; 8.1.52). R. Raab, assr to The Glenn L. Martin Co. Solidified non-explosive hydrocarbon pellets produced by dispersing a liquid hydrocarbon in a sodium alginate composition and thereafter adding the dispersion to aqueous calcium chloride.

U.S.P. 2,581,464 (16.9.47; 8.1.52). J. D. Zech, assr to Devoe & Reynolds. Halogen-containing polyether polyepoxide compositions obtained by partial dehydrohalogenation of polyhalohydrin polyethers of aliphatic polyhydric alcohols containing from 3 to 6 hydroxyl groups.

U.S.P. 2,580,459 (27.11.46; 1.1.52). D. Papa and E. Schwenk, assr sto Schering Corpn. Production of halogenated aryl fatty acids by reacting a phenyl saturated aliphatic carboxylic acid with halogen in the presence of a silver salt of a carboxylic acid.

U.S.P. 2,580,460 (24.5.50; 1.1.52). T. C. Patton and L. J. Jubanowsky, assrs to Baker Castor Oil Co. Heat and light stable vinyl halide polymer compositions comprising the polymer, an ester of 1 : 2-hydroxystearic acid as plasticizer and a rubbery copolymer of 1 : 3-butadiene and acrylonitrile.

U.S.P. 2,580,468 (13.8.49; 1.1.52). F. C. Schafer and E. K. Drechsel, assrs to American Cynamid Co. Preparing aliphatic hydrocarbon *iso*cyanates by pyrolizing the corresponding tri-(aliphatic hydrocarbon) *iso*cyanurate in the presence of HCl.

U.S.P. 2,580,488 (5.1.49; 1.1.52). H. L. Vogl and A. M. Stieber, assrs to S. C. Johnson & Son Inc. A liquid coating composition comprising a vinyl ether polymer having from 14 to 24 C atoms, a resin, and a major proportion of solvent.

U.S.P. 2,580,550 (25.8.49; 1.1.52). A. G. Jelinek, assr to E.I. du Pont de Nemours & Co. Bis(*p*-chlorophenyl) mercaptole acetone, a compound having miticidal activity.

U.S.P. 2,580,730 (4.5.48; 1.1.52). J. M. Church and E. W. Johnson, assrs to Metal & Thermit Corpn. Chlorinated paraffins are stabilized with up to 5% of a butyl, aryl tin compound.

U.S.P. 2,581,640 (23.8.45; 8.1.52). C. A. Fasold and H. W. Greider, assrs to The Phillip Carey Manufg. Co. A solvent-thinned bituminous cementing composition comprising bitumen, mineral filler, and solvent.

U.S.P. 2,581,906 (1.7.48; 8.1.52). G. E. P. Smith, Jr., assr to The Firestone Tyre & Rubber Co. The use of cresylic acid alkylated by introduction of one to two hydrocarbon substituents of 4 to 12 C atoms per cresylic acid mol as antiflex-cracking agents in rubber.

U.S.P. 2,582,037 (12.11.47; 8.1.52), J. B. Hyde, assr to Crown Zellerbach Corpn. Wax paper coating composition comprising micro-crystalline wax, paraffin wax, and up to 10% of a polyethylene of mol. wt. at least 3000.

U.S.P. 2,582,425 (18.5.49; 15.1.52). E. M. Geiser, assr to Universal Oil Products Co. A resinous copolymer composition formed by reacting a vinyl aromatic compound with a coumarone–indene fraction and a material recovered from a hydrocarbon-catalyst sludge in the presence of an acid catalyst.

U.S.P. 2,582,733 (5.11.49; 15.1.52). J. C. Zimmer and G. W. Duncan, assrs to Standard Oil Development Co. A protective composition consisting of 90 to 99% lub oil and 1 to 10% methyl amine petroleum sulphonate.

U.S.P. 2,581,908 (18.2.50; 8.1.52). S. M. Kinzinger, assr to The Firestone Tyre & Rubber Co. Non-discolouring stabilized vinyl chloride resin compositions comprising a vinyl chloride polymer or copolymer containing at least 80% vinyl chloride, cadmium mercaptide, and a urea–formaldehyde or melamine–formaldehyde resin.

R. C. R.

Corrosion

2204. Corrosion control. J. S. Cook. *Oil Gas J.*, 26.5.52, **51** (3), 221.—First of a series on internal and external protection of storage tanks. Describes use of a coal-tar coating, including surface preparation, priming, and primer drying. G. A. C.

2205. Corrosion control. Anon. *Oil Gas J.*, 2.6.52, **51** (4), 117; 9.6.52, **51** (5), 113.—Pt 2 of " How to Protect Storage Tanks " deals briefly with heating and application of coal tar enamels.

Pt 3 deals with preparation of tank, and pneumatic application of concrete mortar (Gunite) on internal surfaces. G. A. C.

2206. Plastic coatings aid refiners in fight against corrosion. F. L. Resen. *Oil Gas J.*, 19.5.52, **51** (12), 124.—The application of a baked phenolic formaldehyde coating which performs satisfactorily in various organic services at temperatures of 225° F and with inorganic acids and salts in most cases is described. Equipment treated includes heat exchangers, plug valves, piping, and tank interiors.

It is stressed that initial preparation of surface before spraying on plastic coat is most important. G. A. C.

2207. Corrosion-resisting nickel alloys and chemical progress. W. Z. Friend and F. L. LaQue. *Industr. Engng Chem.*, May 1952, **44** (5), 965–70.—Corrosion-resisting properties of principal nickel alloys are summarized by reference to uses in specific manufacturing processes. Analyses of alloys are tabulated according to AISI type number.

E. J. C.

2208. What can be done to combat corrosion of refinery tank roofs. H. F. McConomy and J. J. Hur. *Refiner*, May 1952, **31** (5), 124–6. *Mtg API Divn of Refining, San Francisco, May* 1952).—In recent years the cost of repairs to tank roofs at the Philadelphia refinery of the Atlantic Refining Co. has been in excess of $1 million per year. An economic comparison is presented for the replacement of roofs with $\frac{1}{16}$ inch steel plate, $\frac{1}{4}$-inch steel plate, galvanized $\frac{3}{16}$-inch steel plate, $\frac{1}{16}$-inch aluminium plate, and bituminous base and organic resin base coatings on $\frac{3}{16}$-inch steel plate. More consideration should be given to increasing the thickness of steel roofs, and under severe conditions galvanized steel or aluminium may be warranted. A relatively long life of coatings is required to justify their use. A. R. H.

2209. Refinery painting. W. B. Cook. *Corrosion*, 1952, **8** (3), 93–9.—Paint coating systems are described for the various types of surface occurring in oil refineries, including metals, concrete, plaster, wood, and insulating media. The need for adequate surface preparation is stressed. For economic evaluation of a paint system the criterion recommended is cost per unit area per time-unit of durability, based on plant scale applications. K. G. B.

2210. Cathodic protection of buried pipelines—2. K. A. Spencer. *Petroleum*, June 1950, **15** (6), 149–53 (see Abs. 1972).—Pt II describes the type of survey that should be carried out before cathodic protection methods are applied to an existing pipeline, and reviews the protection equipment required, including systems incorporating sacrificial anodes and those supplied with direct current. R. E. P.

Engines and Automotive Equipment

2211. Automobile test rigs. L. H. Dawtrey. *Instn Mech. Engrs (Auto. Div.)*, 1949–50, Pt IV, 176–92.—The background of automobile rig testing and its development from the earlier experimental shop is outlined. Typical test rigs for fatigue failure, opposed frames, spring fixing bolts, laminated springs, saloon bodies shock loading are described, together with impact, transmission, rear axle, and flow rigs. Data are presented on the development of accurate calculation of motor vehicle acceleration and max speed, and it is stressed that whilst rig tests can provide useful and quickly-obtained results, they can in no wise be considered as replacing tests of vehicles on the road. J. G. H.

2212. Performance and weight of automobile petrol engines. D. Bastow. *Proc. Instn Mech. Engrs (Auto. Div.)*, 1950–1, Pt II, 68–81.—Results are presented of a general survey of the influence of such factors as valve arrangement, head design, O.N., and sp. wt., on the power output of automobile engines. Comparisons are made tabulating results obtained with a series of engines each intended to develop 100 b.h.p. J. G. H.

2213. Application of petrol injection to automobile engines. R. Barrington and E. W. Downing. *Proc. Instn Mech. Engrs (Auto. Div.)*, 1949–50, Pt III, 133–54.—Results are reported of an investigation as to whether petrol injection was a worth-while proposition for the ordinary automobile engine. Experiments carried out on four-cyl engines of some 1·5 litres capacity led to the development of a satisfactory system capable of giving equal fuel distribution to all cyl with a tolerance of $\pm 2\%$ at all loads; manifold injection proved superior to direct injection, and it was established that petrol injection of itself will not give any better combustion efficiency than carburation; any saving in the fuel consumption obtained can be due only to better fuel distribution between cyls, better matching of fuel/air ratio to a desired value, or increase in compression ratio made possible by elimination of the hot spot. Under certain conditions resultant saving in part load on bench tests could amount to 10 to 15%. Elimination of the choke and slight increase of C.R. can increase max power by 15 to 20%. Road tests confirmed the results obtained in bench tests and brought out the effects of other factors. It is suggested that an engine fitted with petrol injection is more pleasant to drive by reason of its better idling, more uniform and smoother torque, and increased power at fuel throttle. Results obtained do not bear out some of the more exaggerated claims sometimes made for petrol injection. J. G. H.

2214. Piston assemblies for road transport oil engines. J. L. Hepworth. *Proc. Instn Mech. Engrs (Auto. Div.)*, 1949–50, Pt 11, 69–87.—Frequent types of piston failure, breakage, seizure, ring-sticking, land breaking and scuffing, and groove destruction are noted and measures suggested for their avoidance. Factors limiting piston life as indicated by oil consumption are considered, dangers of overscraping new engines are stressed, and the use of a special scraper ring with delayed action as a means of retaining min oil consumption over a long life is suggested. An extended investigation of rates of wear indicated that the top compression ring is the weakest point in piston assembly. Service tests with piston ring materials showed chromium plated rings to give better results. Suggestions for piston design are presented, with a 12% silicon alloy as material and an adequate path for heat flow from piston crown to ring zone. Design formulæ are presented for gudgeon-pin design. J. G. H.

2215. Significant contribution of the Diesel Research Laboratory. (James Clayton Lecture). C. G. A. Rosen. *Proc. Instn Mech. Engrs (Auto. Div.)*, 1950–51, Pt 1, 7–25.—Diesel engine development work carried out at the Diesel Research Laboratory of the Caterpillar Tractor Co. is reviewed. Intensive study of the combustion process has resulted, after prolonged trials, in the evolution of a modified pre-combustion chamber engine, which is described in some detail together with a discussion of combustion-gas composition, flame duration, temp distribution, the mechanism of ignition, and combustion chamber deposits. The development of fuel injection equipment, pre-calibration, and the evolution of the fuel pump are outlined; the characteristics of several types of check valve and fuel valve are described, and spray characteristics and flow through the fuel valve orifice are examined. Comments are offered on materials for cyl liners and piston rings and on the effect of fuel inclusions on cyl wear. Details are presented of a quartz window designed to facilitate the visual study of combustion phenomena without the distraction of soot condensation on cold windows. J. G. H.

2216. Development of the De Havilland series of engines for light aircraft. J. L. P. Brodie. *Instn Mech. Engrs (Auto. Div.)*, 1950–51, Pt 11, 48–67.—The history of the De Havilland engines up to 1927 is briefly summarized, and the Cirrus and Gypsy engines are described in some detail, followed by a description of the inverted engine introduced in 1931. Other aspects treated include six-cyl engines, variable pitch propellers, torsional vibration, and post-war development. Detailed drawings and comprehensive discussions are included. J. G. H.

Safety Precautions

2217. Security in oilfield practice. K. Krukierek. *Nafta*, 1951, **7**, 54–5.—In Polish petroleum industry the employees are often recruited from peasantry and have to walk a long way to the field. This contributes greatly to tiredness and higher accident rate. Comparisons are made with U.S.S.R., where accidents are on the decline. M. S.

2218. Explosion prevention measures for electrical installations. W. Kobylinski. *Nafta*, 1951, **7**, 51–3; 84–7.—Old Polish regulations dealing with coal mines and oil wells are not adequate for oil refineries. Russian and German codes are quoted. These divide gases and vapours according to their volatility, explosiveness, and toxicity. Different precautions are necessary for different groups of gases. French and British regulations are reviewed, and in conclusion author expresses himself in favour of the first-named two, provided numerical limits of gas content can be incorporated in these regulations. M. S.

Miscellaneous

2219. Petroleum and the textile industry. F. N. McLaren. *IP Review*, 1952, **6**, 87.—A brief history of the different branches of the textile industry is given. Products from the petroleum industry are used in a number of processes, these uses being covered by three main headings: fuels, lubricants, and processing materials. Each group is

dealt with in turn. Many engines have now been converted to oil-firing types, and the consumption of diesel oil is expected to increase. The engines involved in the textile industry have generally high speeds and light loads, and the lubrication process is a simple one. However, there are difficulties due to staining of the fabric by the lubricant, and attempts to solve this problem are of both the " prevention " and " cure " kind, involving the use of tacky oils and emulsifiable oils. Processing involves oils used as " wool oils " for fibre lubrication, for dust suppression in the carding of cotton, as fibre lubricants and delustrants in the rayon industry, as detergents, and for solvent degreasing of wool. R. H.

2220. Carbon black production. Anon. *India-Rubber J.*, 1951, **121**, 440.—Estimates are given from the National Production Authority for U.S. carbon black production capacity and demand for 1951 and 1952. A substantial excess of output over domestic demand in 1952 will permit more exports. C. N. T.

2221. Industrial and technical planning in petroleum industry. S. Laskowski. *Nafta*, 1951, **7**, 26–8.—It is suggested that plans for refining must in the first place depend on plans for drilling, and thus a form of a ladder is obtained. Each " rung " is made on the basis of statistical data. M. S.

2222. Progress in the field of petroleum chemistry. K. Baranski. *Nafta*, 1951, **7**, 48–51; 81–4.—Deals with advances in refining processes, *e.g.*, extraction by means of solvents, azeotropic distillation, superfractionation. Then cracking, dehydrogenation, pyrolysis, isomerization alkylation, hydroforming, and catalytic processes are described; and deals with separation of unsaturated products. Short description is given of the manufacture of chlorosulphones and detergents. M. S.

2223. Science and technique in Soviet Union (Monthly Review). Anon. *Nafta*, 1951, **7**, 55–7.—Deals with: (1) history of progress of the Soviet petroleum industry from drilling to refining; (2) " Electro-Vert " unit of 4000 kg wt, 100 to 125 kW; 1500 r.p.m. consisting of motor, reducing gears, and drilling pipe to be introduced into the bottom of the well; (3) flameless gas burners for low cal. val. gases. Gas and air burn in a tunnel leading to outlet, and exhaust gases contain no oxygen or CO. M. S.

2224. Progress in domestic heating research. A. C. Monkhouse. *J. Inst. Fuel*, May 1952, **25** (143), 103.—A historical survey of domestic heating is given. Progress in the testing of appliances is described, with reference to the measurement of radiation, convection, water heating and air flow, and to the contributions made by research organizations and industry. Two types of test are distinguished: (*a*) the appliance as a producer of heat; (*b*) the appliance in relation to the house. The influence of insulation of houses and research on different types of heating appliances in houses with the same standard of insulation was investigated by the Building Research Station of the D.S.I.R. Research was also carried out on the effect of environment on physiological and comfort conditions. The prevention of smoke and the production of smokeless solid fuels is considered. District heating, either by providing hot water from a central boiler, or utilizing waste heat of power stations, and the heat pump are stated to have only limited application for domestic use at the present time.

The aims of research are: to provide house heating and hot water to modern standards; to use coal more efficiently, thereby reducing its consumption; to decrease atmospheric pollution by smoke abatement. D. K.

2225. Investigation of whole-house heating. F. C. Lant and J. C. Weston. *J. Inst. Fuel*, May 1952, **25** (143), 109.—Temp, fuel consumption, and expenditure for houses with various types of whole-house heating are given and compared with data for conventional heating arrangements. Studies of temp gradients occurring with different heating arrangements were made at most of the sites. Results obtained suggest that for open-plan houses, heating by warm air from a low-level inlet on the ground floor, possibly with a simple register in a small house, provides the best temp distribution. The effects of seasonal variation of temp and fuel consumption are discussed. D. K.

2226. Presidential Address, 1952. Fuel technology and civilization. G. E. Foxwell. *J. Inst. Fuel*, May 1952, **25** (143), 86.—The aim is to show that though fuel technologists must occupy themselves with urgent current problems, they should not neglect the wider implications of the work or the future social problems that will concern the profession.

The history of fuel application from pre-Roman times through the era of the industrial revolution is reviewed. The influence of fuel development on social conditions is particularly noted.

The present day fuel problems of both Britain and the world and the future outlook are discussed. D. K.

2227. Technical and hygienic requirements for solvents for preservatives for structures. A. W. Rick. *Bitumen, Teere, Asphalte, Peche*, 1952, **1** (3), 17–20.—Due to anxiety regarding the market for solvents for preservatives for structures, a brief discussion on the production, uses, and properties of the most important types is presented. Further, the sphere of application in question and the grounds for limiting their use or rejecting them are discussed. Normally, solutions on a tar or bitumen basis, also heavy benzole and white spirits may be used as solvents for building preservatives. From technical and industrial hygiene viewpoints precautionary measures are to be taken in the working up and use of all other solvents. R. T.

BOOK REVIEW

The Performance of Lubricating Oils. H. H. Zuidema. New York: Reinhold Publishing Corp.; London: Chapman & Hall, pp. 179 + viii, 40*s*.

This is one of the monograph series that have been produced by The American Chemical Society at intervals since 1921. As a monograph it should survey thoroughly some section—albeit a restricted one—of science or technology, and this book adequately fulfils this purpose.

The author discusses lubricants from the viewpoint of their behaviour under the conditions of their use. This is a more fundamental approach than that usually made, where the properties of an oil are considered relative to a specific application. The subject matter is covered in eight chapters, each provided with a good bibliography.

The opening chapter discusses lubrication under hydrodynamic and boundary conditions, and more or less follows the conventional treatment of this subject. This leads on to the consideration of extreme pressure lubricants and a description of the various types of test apparatus that have been developed to rate this property. Chapter II is devoted to a dissertation on rheology—a subject which has received its full share of attention during recent years. Much has been written on the rheological properties of lubricants, and it would be impossible to cover the whole field without endangering the balance of the book. The author's survey is, however, adequate and reasonably up to date. The chapter includes sections on pour point depressants, viscosity index and V.I. improvers, and viscosity/pressure relationships.

Oxidation forms the subject of Chapter III, and is dealt with under two heads—oxidation products and oxidation rates. The oxidation of single hydrocarbons is first considered, and the hydroperoxide theory is discussed at length. Then follows a treatment of mixtures and petroleum fractions and the work of Dornte, Fenske, and others on oxygen absorption measurements. The question of oxidation rates brings in the subject of catalysts and inhibitors, a very wide field, which is covered in five pages.

Three chapters treat of matters of great importance in the use of lubricants in engines—Chapter IV, Bearing corrosion; Chapter V, Sludge and lacquer deposition; and Chapter VII, Wear. Again within the confines of this book the subjects are well presented.

Emulsification and foaming form the subject of Chapter VI, and the theory of emulsions is discussed before the practical aspects, as instanced by steam turbine oils and soluble cutting oils, are considered. The formation of foams does not appear to have been satisfactorily explained, and much of the work on foaming concerns boiler water problems. The use of silicones to suppress foaming in oils has become almost universal, and appears to depend on their insolubility in the lubricant. A brief final chapter (Chapter VIII) deals with manufacturing methods and includes references to synthetic and non-hydrocarbon oils.

Although, as is to be expected, the majority of the references in this book are to American publications, the author has given due consideration to work done outside the U.S.A. and the bibliographies include many British references.

The book is somewhat uneven, being scrappy and disjointed in parts, and does not always read smoothly. The notation is also somewhat confusing in places. Thus in the chapter on Rheology P signifies density and later on pressure, while V is made to represent volume and viscosity. The definition of stokes as poises/density is not good following a comparison of English and metric units, and would have been better expressed as poises/specific gravity.

For the rest the book is well printed on good paper and contains a number of good diagrams, graphs, and pictures. It will be extremely useful to the lubricant man, to whom it can be recommended without hesitation. D. L. S.

ADDITIONS TO THE LIBRARY

Wax Chemistry and Technology. L. Ivanovszky. London and Bridgend: The Author, 1952. Pp. 44.

Reprint of a paper first published in four instalments in *Soap, Perfumery, and Cosmetics* in 1951–52. In three parts, the first part discusses waxes as a specific group, the second deals with chemistry, physics, and analysis, while the final part is devoted to the individual waxes.

Measurement of Gas Law Deviations with Bean and Burnett Apparatus. O. T. Bloomer. Chicago, Ill.: Institute of Gas Technology, 1952. Pp. 12. $2.00.

A comprehensive study of use of the Bean and Burnett apparatus and of its accuracy and reproducibility in determinations of deviations from the ideal gas law. This Research Bulletin No. 13 also serves as an instruction manual and guide to the use of the two apparatuses.

Physical-Chemical Properties of Methane–Nitrogen Mixtures. O. T. Bloomer and J. D. Parent. Chicago, Ill.: Institute of Gas Technology, 1952. Pp. 35. $3.50.

This Research Bulletin No. 17 of the IGT contains a literature survey, and details of apparatus designed for and the experimental techniques used in the determination of the phase equilibrium and p-v-t relationships for methane, nitrogen, and mixtures thereof.

THE INSTITUTE OF PETROLEUM

NOTICES, AUGUST 1952

ENTRANCE FEES AND SUBSCRIPTIONS

The current Entrance Fees and Annual Subscriptions for membership of the Institute are :

	Entrance Fee	*Annual Subscription*
	£ *s.* *d.*	£ *s.* *d.*
Members and Fellows	2 2 0	4 4 0
Associate Members and Associate Fellows .	1 1 0	3 3 0
Student Members	Nil	1 1 0

APPLICATIONS FOR MEMBERSHIP OR TRANSFER

The following have applied for admission to or transfer of membership in the Institute. In accordance with the By-laws, the proposals will not be considered until the lapse of at least one month after the publication of this *Journal*, during which time any Fellow, Member, Associate Fellow, or Associate Member may communicate by letter to the Secretary, for the confidential information of the Council, any particulars he may possess respecting the qualifications or suitability of the candidate.

The object of this information is to assist the Council in grading the candidate according to the class of membership.

The name of candidates' proposers and seconders are given in parentheses.

Application for Membership

BARKHURST, Carlton Raymond, refinery manager, Bahrein Petroleum Co. (*R. Godfrey; N. E. Watson.*)

BATTEN, Leonard Ralph, manager (Patent Licence Department), Shell Petroleum Co., Ltd. (*M. A. Matthews; W. F. Mitchell.*)

BEATTIE, Robert Dudley, acting superintendent of refinery, National Oil (Pty) Ltd. (*G. W. E. Mapstone; G. Sell.*)

CANNON, John James, design engineer, Lummus Co. Ltd. (*T. Shelley; J. E. Jones.*)

DUNCAN, Stanley Brownlie, manager (Petroleum Chemicals Department), Cyanamid Products Ltd. (*G. H. Thornley; J. C. Cragg.*)

DUNLOP, Robert Woodhill, works manager, Marks & Johnston Ltd. (*H. MacArthur; W. M. Cumming.*)

GLASS, George Paxton, economist (Economic Research Section), Shell Petroleum Co. Ltd. (*O. F. Thompson; T. C. Outram.*)

HEAP, Harry Wells, London representative, Bechtel International Corp. (*C. A. P. Southwell; E. C. Masterson.*)

HUGHES, John Richard, safety engineer, Anglo-Iranian Oil Co. Ltd. (*C. S. Cleverly; F. A. Maliphant.*)

JOHNS, Thomas Clifford, student apprentice, National Oil Refinery Ltd. (*P. F. Ellis; E. J. Horley.*)

KING, Francis James, project manager (refinery projects), Kellogg International Corp. (*J. E. Lytle; G. E. Wheeler.*)

MUMFORD, Peter Edward Brewster, technical inspector, Ministries of Supply, Works & Pensions. (*E. C. Masterson; G. Sell.*)

STEEL, Howard James, sales representative, Shell-Mex & B.P. Ltd. (*A. Moon; G. Sell.*)

WALTON, Hereward Royden, sales manager, Fisher Governor Co. (*S. W. I. Wallis; C. R. Sams.*)

WHINERAY, John Douglas, consulting engineer and surveyor, Messrs. Carmichael & Clarke. (*A. W. Black; W. Wright.*)

WOOD, Frederick William, topping plant shift operator, Kuwait Oil Co. Ltd. (*R. L. Wilson; K. Turner.*)

Transfers

GRIFFITHS, Thomas Haydn, assistant mechanical engineer, Anglo-Iranian Oil Co. Ltd. (*P. F. Ellis; E. J. Horley.*) (*Student to Associate Fellow.*)

RAFFERTY, Joseph, workshop time-clerk, Rootes Ltd. (*E. W. E. Dalling; H. F. Porter.*) (*Student to Associate Fellow.*)

NEW MEMBERS

The following elections have been made by the Council in accordance with the By-laws.

Elections are subject to confirmation in accordance with the By-laws.

As Fellow

MITCHELL, W. F.

Transferred from Member to Fellow

TAIT, E. J. H.

As Members

ESCOTT, W. A. T.
FROST, V. H.
KNOTT, C.
KOELMAN, A. D.

As Associate Fellows

BARK, W. R.
BLACKWOOD, R. K.
BONES, D. E.
CAESAR, J.
CONNOLLY, I. D. C.
DAVEY, P. P.
EDLER, E.
GRAYSTON, G. W.
HAY, K. G. B.
HENRY, K. G.
HARRAP, G. N.
HARGREAVES, E.
KITCHIN, P. J.
LITTLEWOOD, A. J.
RISSON, A. E.
RUSH, E. A. E.
SHEASBY, G. R.
SMEDLEY, L. J.
TERRINGTON, A. E.
TRAYLEN, G. J.
VERGHESE, J.
WILSON, K.
WHOLEY, M. W.
WATSON, D.

Transferred from Students to Associate Fellows

BEXON, R.
SHEASBY, G. R.

As Associate Members

HERRERA, D. A.
WILLIAMS, L. H.
MARTIN, C. S.

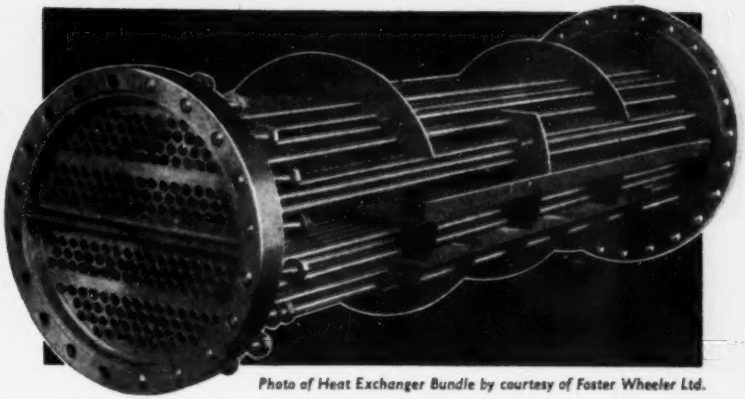
As Students

BENNETT, J. A. R.
COLEMAN, M. C.
KING, P. J.

CORRECTION

MARTIN, Cyril Stanley. In the *Journal* for June 1952 Mr Martin should have been described as "manager, White Products (General), London Div'n, Shell-Mex & B.P. Ltd."

True to Specification . . .



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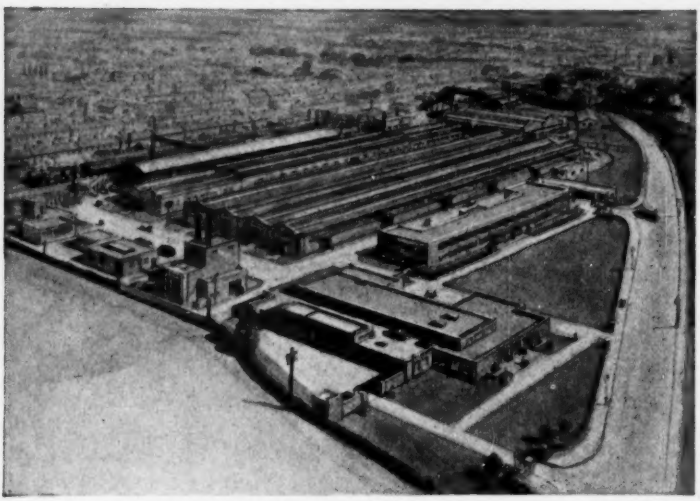

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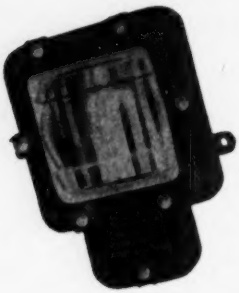

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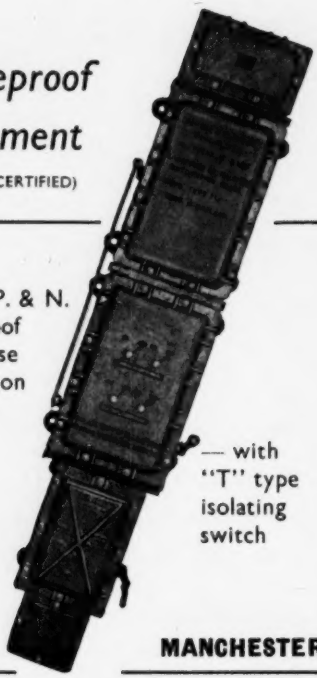

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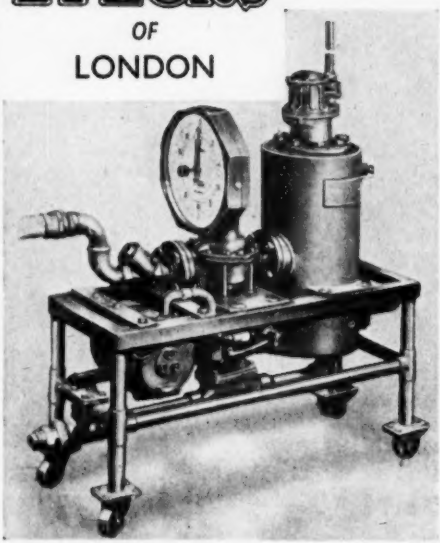

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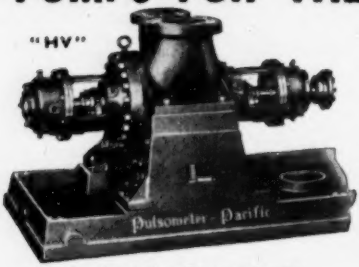

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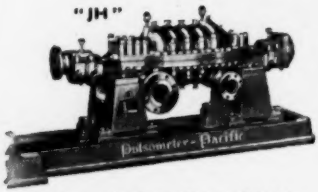

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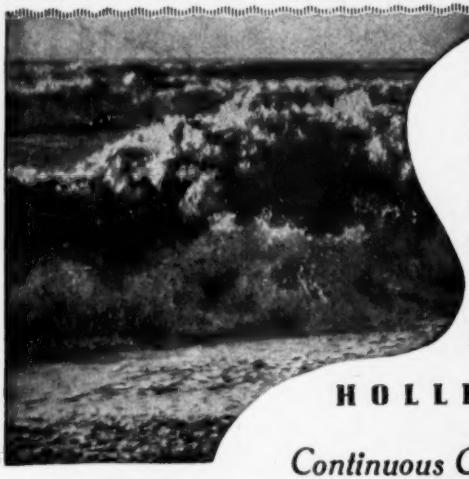

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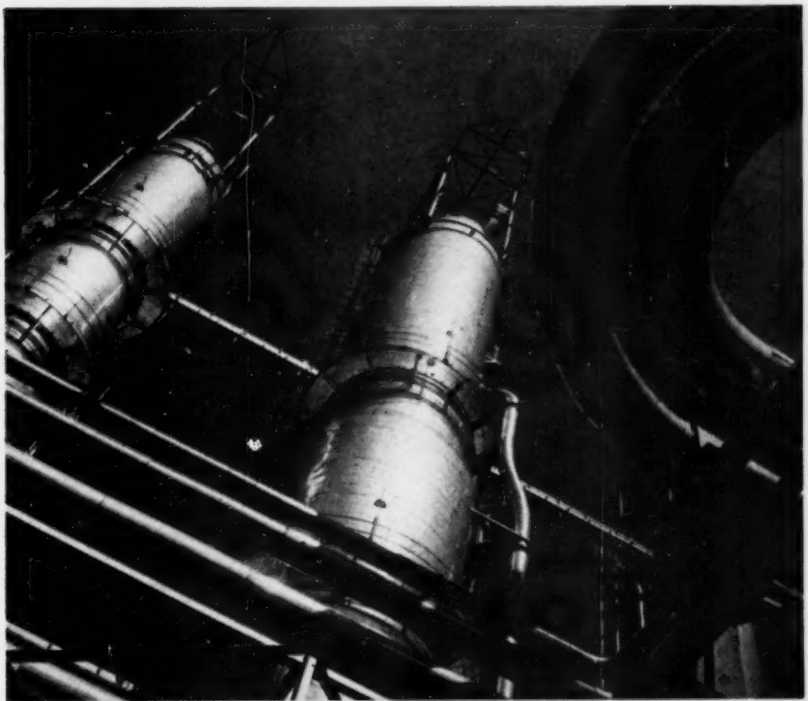

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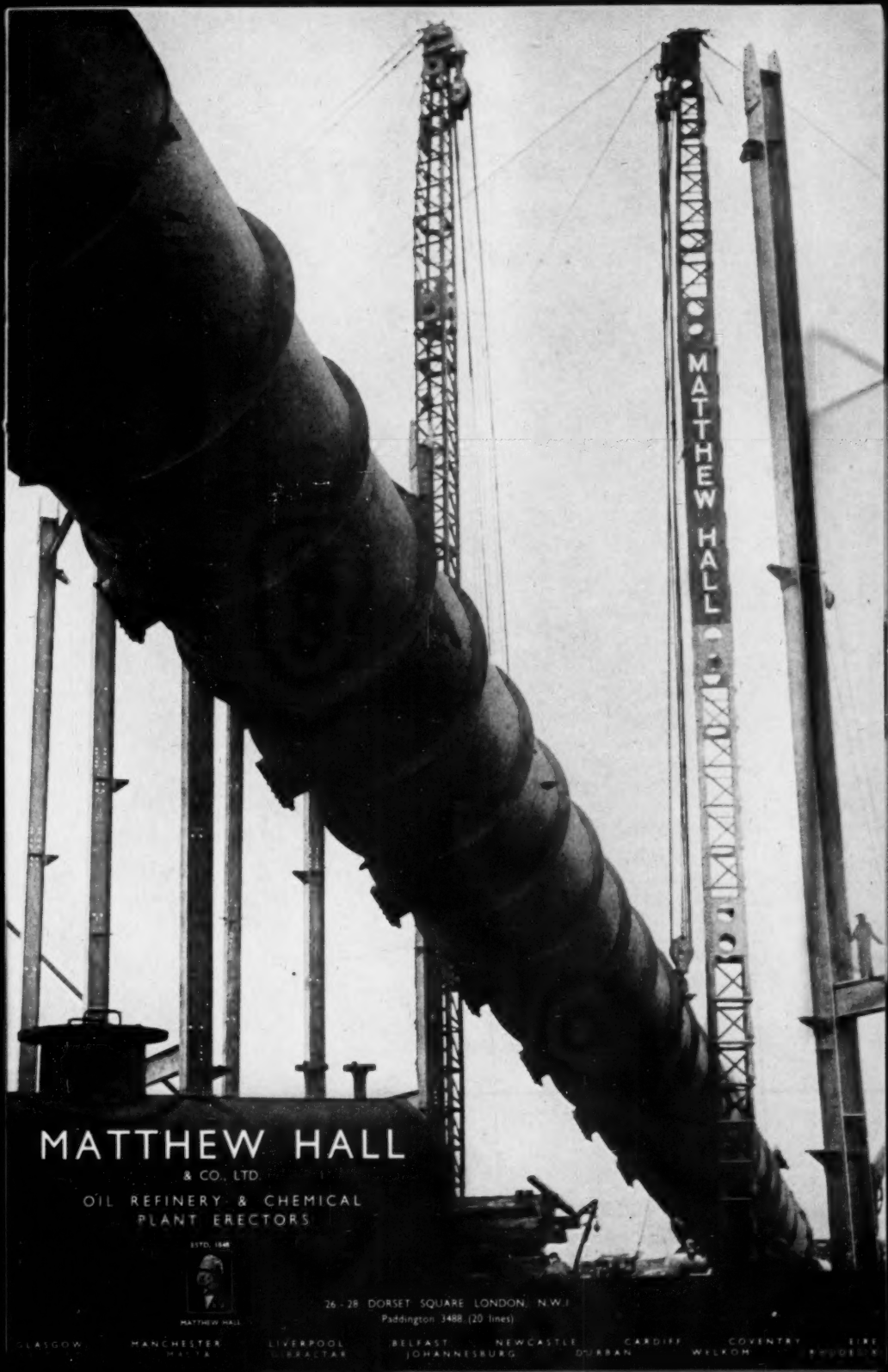

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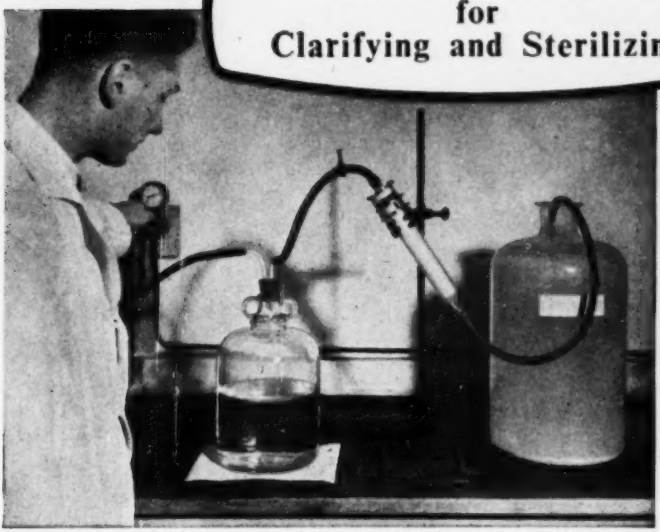

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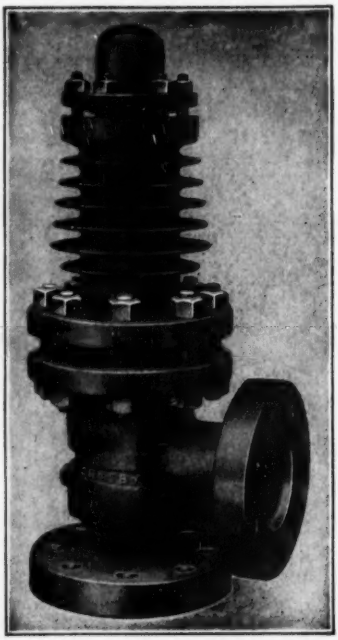
Royal DOULTON FILTERS



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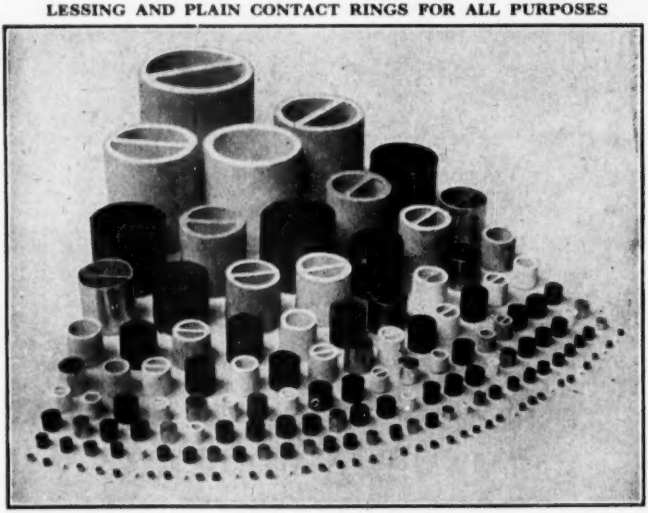

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This Structure at the Vacuum Oil Company's new refinery at Coryton Essex, will form the basis for distillation and fractionating columns. It is being mechanically erected by Costain-John Brown Ltd., and engineered by The Lummus Company. Fleetweld 5 electrodes and Lincoln machines are being used, of which two SAE. 300 Motor Generators can be seen on the job.

'We ought to ask Lincoln before we decide'

World's largest manufacturers of arc-welding equipment and electrodes LINCOLN

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THE DANGER OF A SAFETY VALVE failing to operate is only too evident, particularly if it is fitted to a vessel containing corrosive liquor under pressure. That is why vital parts of the 'Sergeant' safety valve are made of Monel.

THE SAFETY OF MONEL lies in its resistance to attack by many corrosives, combined with strength and toughness. The use of Monel* ensures reliability even when safety valves are left without much attention for long periods.

Although Monel, like other Wiggin nickel alloys, is at present in short supply, the time will come when designers of industrial plant will once again be able to use it wherever they need a strong, tough material to resist corrosion.

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Speedily and without fail Nicerol Foam KILLS petrol and oil fires—and is used by the Royal Navy, Royal Air Force and Oil Companies specifically for that purpose. Nicerol is a concentrate producing a **heavy type of vapour-sealing foam** which flows with creeping flame, blanketing and killing the fire—resisting effectively all possible flash-back or re-ignition. Any type of mechanical or air-foam apparatus will produce foam from Nicerol—either with **fresh** or **salt** water.

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Nicerol is supplied in new heavy gauge steel drums specially designed to permit quick release and clean pouring of contents. These are supplied in 2, 5 or 40 Imperial gallons capacity. 3 gallons of Nicerol will produce over 1,100 gallons of pure white foam.

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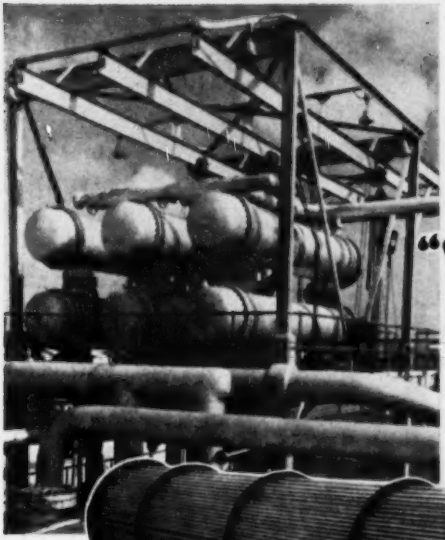

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FOR
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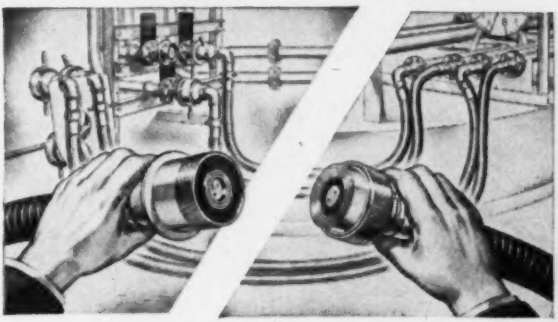
BREAK . . .
without
spilling . . .



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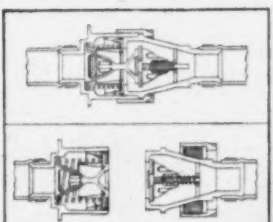

There is an automatic valve in each half of an Exactor Coupling. When the coupling is joined, the valves mate and keep open by mutual pressure. When you uncouple, each valve seals its side of the break. Full range of sizes from ¼" to 7" bore, for high and low pressures.

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Seal as they break

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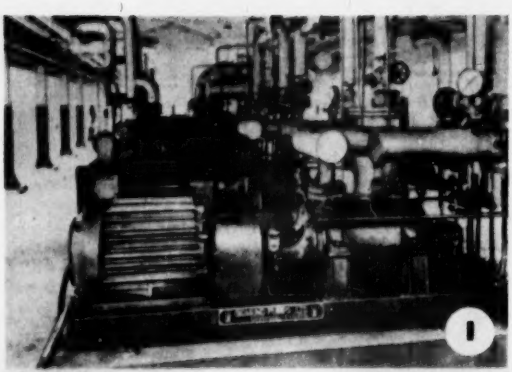

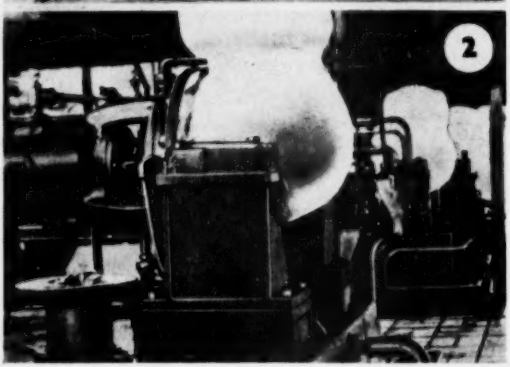

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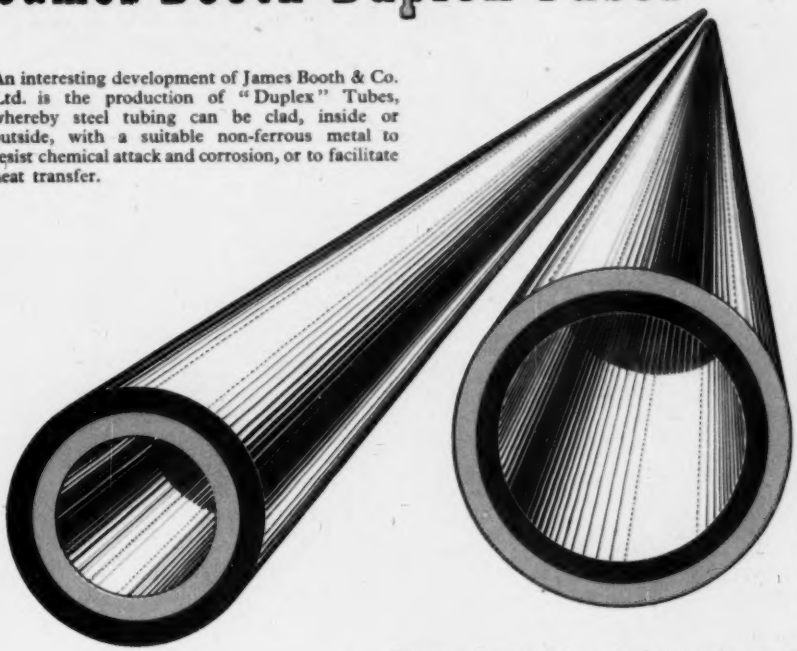

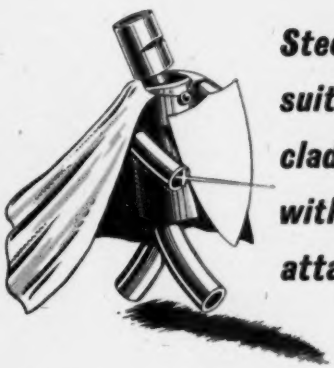

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The corrosion of plain steel tubes may be caused by the presence or passage of corrosive substances through or around the tubes, and may be particularly severe where the tubes are alternatively wet and dry as a result of the use of cooling media. Such attack may be severest either inside or outside the tube, depending upon the specific conditions of its use. The life of tubing in cracking, fractionating or heat exchanging plant can therefore be extended to an enormous degree by the use of "Duplex" Tubes.

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BOOTH BIRMINGHAM

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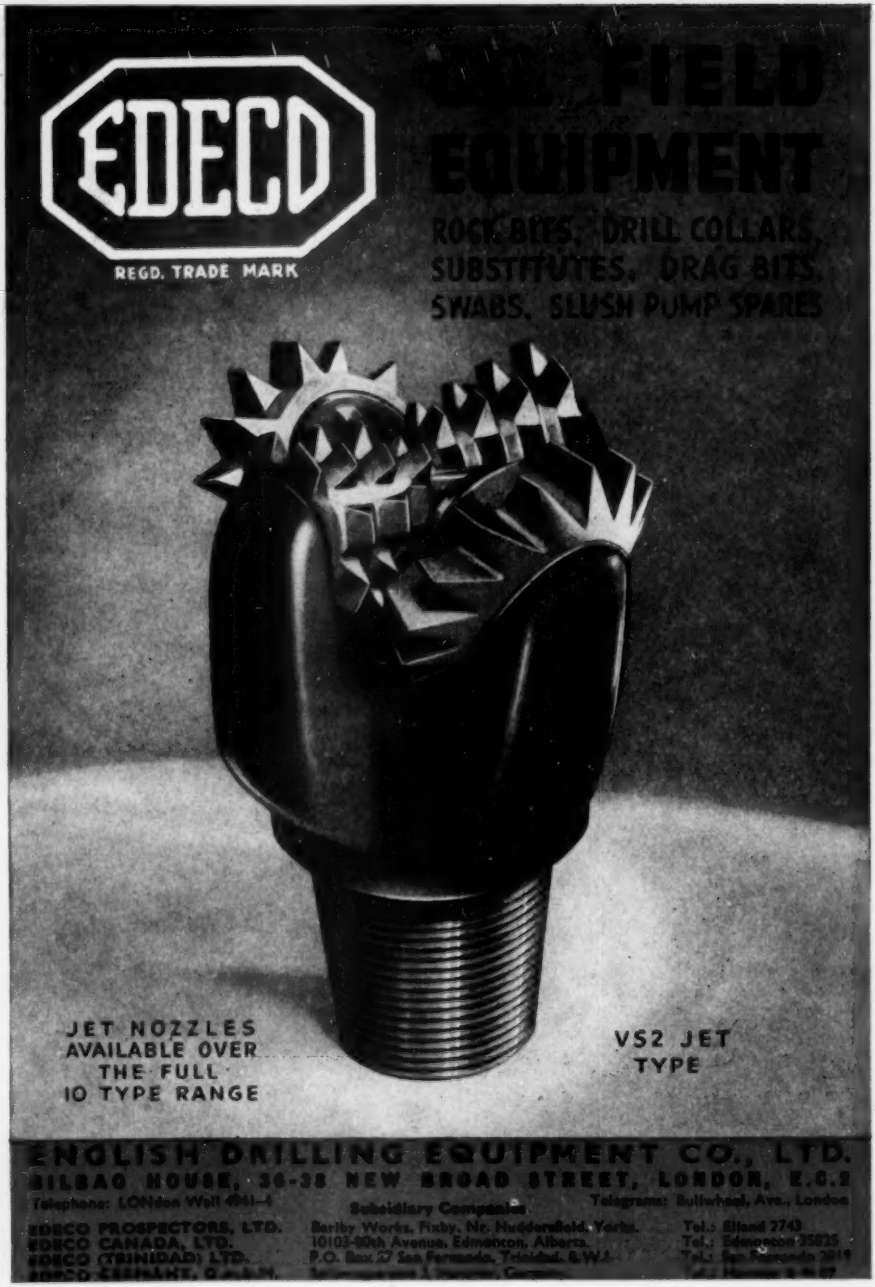
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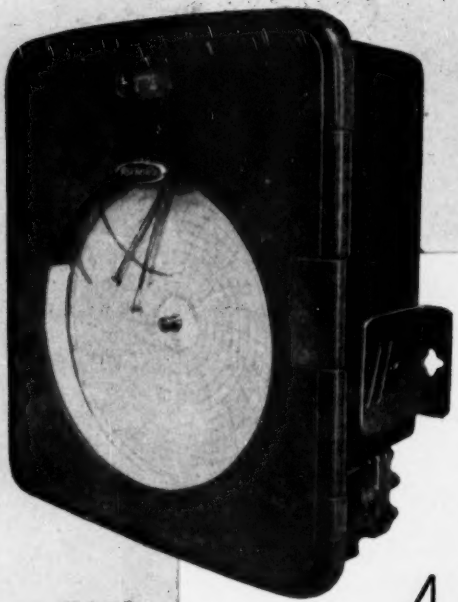

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